THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

# TRANSACTIONS

## VOLUME 35

BALTIMORE MEETING
NEW YORK MEETING
1913

NEW YORK
PUBLISHED BY THE SOCIETY
29 West 39th Street
1914

# OFFICERS
# THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
## FORMING THE STATUTORY COUNCIL
## 1913

### *PRESIDENT*

W. F. M. Goss .......... Chicago, Ill.

### *VICE-PRESIDENTS*

Wm. F. Durand .......... Palo Alto, Cal.
Ira N. Hollis .......... Cambridge, Mass.
Thos. B. Stearns .......... Denver, Colo.

Terms expire at Annual Meeting of 1913

James Hartness .......... Springfield, Vt.
Irving E. Moultrop .......... Boston, Mass.
Henry G. Stott .......... New York City

Terms expire at Annual Meeting of 1914

### *PAST-PRESIDENTS*

Members of the Council for 1913

M. L. Holman .......... St. Louis, Mo.
Jesse M. Smith .......... New York City
George Westinghouse .......... Pittsburgh, Pa.
E. D. Meier .......... New York City
Alex. C. Humphreys .......... New York City

### *MANAGERS*

David Francis Crawford .......... Pittsburgh, Pa.
Stanley G. Flagg, Jr. .......... Philadelphia, Pa.
Edwin Britton Katte .......... New York City

Terms expire at Annual Meeting of 1913

Chas. J. Davidson .......... Chicago, Ill.
Henry Hess .......... Philadelphia, Pa.
George A. Orrok .......... New York City

Terms expire at Annual Meeting of 1914

Wm. B. Jackson .......... Chicago, Ill.
Henry Martin Leland .......... Detroit, Mich.
Alfred Noble .......... New York City

Terms expire at Annual Meeting of 1915

### *TREASURER*

Wm. H. Wiley .......... New York City

### *CHAIRMAN OF THE FINANCE COMMITTEE*

Robert M. Dixon .......... New York City

### *HONORARY SECRETARY*

Frederick R. Hutton .......... New York City

### *SECRETARY*

Calvin W. Rice .......... 29 West 39th Street, New York City

# PAST OFFICERS

Alexander L. Holley, Chairman of the Meeting for Organization of The American Society of Mechanical Engineers........Died January 29, 1882

## *PRESIDENTS*

| | | |
|---|---|---|
| R. H. Thurston | 1880-1882 | Died October 25, 1903 |
| E. D. Leavitt | 1883 | Cambridge, Mass. |
| John E. Sweet | 1884 | Syracuse, N. Y. |
| J. F. Holloway | 1885 | Died September 1, 1896 |
| Coleman Sellers | 1886 | Died December 28, 1907 |
| George H. Babcock | 1887 | Died December 16, 1893 |
| Horace See | 1888 | Died December 14, 1909 |
| Henry R. Towne | 1889 | New York City |
| Oberlin Smith | 1890 | Bridgeton, N. J. |
| Robert W. Hunt | 1891 | Chicago, Ill. |
| Charles H. Loring | 1892 | Died February 5, 1907 |
| Eckley B. Cox | 1892-1894 | Died May 13, 1895 |
| E. F. C. Davis | 1895 | Died August 6, 1895 |
| Charles E. Billings | 1895 | Hartford, Conn. |
| John Fritz | 1896 | Died February 13, 1913 |
| Worcester R. Warner | 1897 | Cleveland, Ohio |
| Charles Wallace Hunt | 1898 | Died March 27, 1911 |
| George W. Melville | 1899 | Died March 17, 1912 |
| Charles H. Morgan | 1900 | Died January 10, 1911 |
| S. T. Wellman | 1901 | Cleveland, Ohio |
| Edwin Reynolds | 1902 | Died February 19, 1909 |
| James M. Dodge | 1903 | Philadelphia, Pa. |
| Ambrose Swasey | 1904 | Cleveland, Ohio |
| John R. Freeman | 1905 | Providence, R. I. |
| F. W. Taylor | 1906 | Philadelphia, Pa. |
| F. R. Hutton | 1907 | New York City |
| M. L. Holman | 1908 | St. Louis, Mo. |
| Jesse M. Smith | 1909 | New York City |
| George Westinghouse | 1910 | Pittsburgh, Pa. |
| E. D. Meier | 1911 | New York City |
| Alex. C. Humphreys | 1912 | New York City |

## *HONORARY COUNCILLORS*

1913

| | | |
|---|---|---|
| M. L. Holman | 1908 | St. Louis, Mo. |
| Jesse M. Smith | 1909 | New York City |
| George Westinghouse | 1910 | Pittsburgh, Pa. |
| E. D. Meier | 1911 | New York City |
| Alex. C. Humphreys | 1912 | New York City |

**According to the Constitution, Article C 27, the five Past-Presidents who last held the office shall be members of the Council, with all the rights, privileges and duties of the other members of the Council.**

# EXECUTIVE COMMITTEE OF THE COUNCIL

W. F. M. GOSS, *Chairman*
ALEX. C. HUMPHREYS, *Vice-Chairman*
EDWIN BRITTON KATTE
E. D. MEIER
GEO. A. ORROK
HENRY G. STOTT

# STANDING COMMITTEES

### *FINANCE*

ROBERT M. DIXON (1), *Chairman*
WALDO H. MARSHALL (2)
HENRY L. DOHERTY (3)
WILLIAM LAWRENCE SAUNDERS (4)
WILLIAM DURHAM SARGENT (5)

### *HOUSE*

EDWARD VAN WINKLE (1), *Chairman*
HENRY R. COBLEIGH (2)
S. D. COLLETT (3)
WILLIAM NOBLE DICKINSON (4)
FREDERICK A. SCHEFFLER (5)

### *LIBRARY*

LEONARD WALDO (4), *Chairman*
CHARLES L. CLARKE (1)
ALFRED NOBLE (2)
WALTER M. MCFARLAND (3)
The Secretary

### *MEETINGS*

H. DE B. PARSONS (1), *Chairman*
LEON PRATT ALFORD (2)
HENRY E. LONGWELL (3)
HENRY L. GANTT (4)
ROBERT H. FERNALD (5)

### *MEMBERSHIP*

HOSEA WEBSTER (1), *Chairman*
THEODORE STEBBINS (2)
WM. H. BOEHM (3)
HENRY C. MEYER, JR. (4)
L. R. POMEROY (5)

### *PUBLICATION*

GEO. I. ROCKWOOD (1), *Chairman*
GEO. M. BASFORD (2)
CHAS. I. EARLL (3)
IRVING E. MOULTROP (4)
FRED. R. LOW (5)

### *PUBLIC RELATIONS*

JAMES M. DODGE (3), *Chairman*
JOHN W. LIEB, JR. (1)
FRED J. MILLER (2)
WORCESTER R. WARNER (4)
GEO. M. BRILL (5)

### *RESEARCH*

RICHARD H. RICE (4), *Chairman*
LIONEL S. MARKS (1)
ADOLPH L. DE LEEUW (2)
ROLLA C. CARPENTER (3)
RALPH D. MERSHON (5)

NOTE:—Numbers in parentheses indicate number of years the member has yet to serve.

## SOCIETY REPRESENTATIVES

*AMERICAN ASSOCIATION FOR ADVANCEMENT OF SCIENCE*

Alex. C. Humphreys — Wm. B. Jackson

*ENGINEERING EDUCATION*

Alex. C. Humphreys — Frederick W. Taylor

*GENERAL CONFERENCE COMMITTEE OF NATIONAL ENGINEERING SOCIETIES*

Charles Whiting Baker — E. D. Meier

*JOHN FRITZ MEDAL*

Henry R. Towne (1) — Frederick R. Hutton (3)
John A. Brashear (2) — John R. Freeman (4)

*TRUSTEES UNITED ENGINEERING SOCIETY*

Alex. C. Humphreys (1) — Fred J. Miller (2)
Jesse M. Smith (3)

Note:—Numbers in parentheses indicate number of years the member has yet to serve.

# GEOGRAPHICAL SECTIONS OF THE SOCIETY

### *ATLANTA*

J. S. Coon, *Chairman*
Park A. Dallis, *Secretary*
George Hillyer, Jr.
Cecil P. Poole
Earl F. Scott

### *BOSTON*

Henry Bartlett, *Chairman*
Ralph E. Curtis, *Secretary*
Herbert N. Dawes
Wm. Gage Snow
Arthur L. Williston

### *CHICAGO*

Paul P. Bird, *Chairman*
Chas. Wm. Naylor, *Secretary*
Charles Robt. Birdsey
Wm. B. Jackson
Arthur Willett Moseley

### *CINCINNATI*

A. L. De Leeuw, *Chairman*
James B. Stanwood, *Vice-Chairman*
John T. Faig, *Secretary*
Walter G. Franz
Geo. W. Galbraith

### *MILWAUKEE*

Fred. H. Dorner, *Chairman*
Matthias A. Beck
Arthur Simon
Henry Weickel
Euclid P. Worden

### *NEW HAVEN*

Erwin S. Cooley, *Chairman*
Edwin H. Lockwood, *Secretary*
Frank L. Bigelow
L. P. Breckenridge
Henry Bradford Sargent

### *NEW YORK*

Frederick A. Waldron, *Chairman*
Edward Van Winkle, *Secretary*
Roy V. Wright, *Treasurer*
Henry R. Cobleigh
John Joseph Swan

### *PHILADELPHIA*

D. Robert Yarnall, *Chairman*
William R. Jones, *Secretary*
Jas. Edwin Gibson
Lewis H. Kenney
Thomas C. McBride

### *MINNEAPOLIS-ST. PAUL*

Max Toltz, *Chairman*
Chas. L. Pillsbury, *Vice-Chairman*
Emil Jennings Heinen

### *ST. LOUIS*

Ernest L. Ohle, *Chairman*
Frederick E. Bausch, *Secretary*
Frank N. Jewett
John Hunter
Louis C. Nordmeyer

### *SAN FRANCISCO*

Andrew M. Hunt, *Chairman*
T. W. Ransom, *Secretary*
Wm. F. Durand
Edward C. Jones
Thomas Morrin

# SUB-COMMITTEES OF THE COMMITTEE ON MEETINGS

### *ADMINISTRATION*

James M. Dodge, *Chairman*
L. P. Alford, *Secretary*
Daniel Moore Bates
John Calder
Holden A. Evans
James Hartness
W. B. Tardy
Alexander Taylor
H. H. Vaughan

### *AIR MACHINERY*

Frederick Wm. O'Neil, *Chairman*
Hugh V. Conrad
William Prellwitz
Richard H. Rice

### *CEMENT MANUFACTURE*

John Gosta Bergquist, *Chairman*
H. J. Seaman, *Vice-Chairman*
Gabriel Scott Brown
William Robert Dunn
Frederick Williams Kelley
Morris Kind
Frederick Humphreville Lewis
William H. Mason
Richard Kidder Meade
Ejnar Posselt
Holger Struckmann
Arthur C. Tagge
Percy Hartshorne Wilson

### *DEPRECIATION AND OBSOLESCENCE*

Alex. C. Humphreys, *Chairman*
John Gosta Bergquist
Charles Jackson Davidson
Alfred E. Forstall
Frederick Williams Kelley
Holger Struckmann

### *FIRE PROTECTION*

John R. Freeman, *Chairman*
Edw. Vinton French, *Vice-Chairman*
Albert Blauvelt
F. M. Griswold
Holbrook F. J. Porter
T. W. Ransom
Ira. H. Woolson

### *HOISTING AND CONVEYING*

Richard B. Sheridan, *Chairman*
C. Kemble Baldwin
Alex. C. Brown
Orton G. Dale
P. J. Fickinger
Frank E. Hulett
Spencer Miller
A. L. Roberts
Harry Sawyer

### *INDUSTRIAL BUILDING*

Frederick A. Waldron, *Chairman*
Harry A. Burnham
Charles Day
William Dalton
John O. DeWolf
Charles T. Main

### *IRON AND STEEL*

Joseph Morgan, *Chairman*
Thomas Towne, *Secretary*
W. P. Barba
Frank F. Beall
Rogers Birnie
Albert Ladd Colby
Julian Kennedy
Marcus T. Lothrop
William E. Snyder
James T. Wallis
R. M. Watt

### *MACHINE SHOP PRACTICE*

FRED H. COLVIN, *Chairman*
EDWARD P. BULLARD, JR.
LUTHER D. BURLINGAME
WALTER L. CLARK
ADOLPH L. DE LEEUW
WILLIS H. DIEFENDORF
FREDERICK L. EBERHARDT
FRANKLIN A. ERRINGTON
ARTHUR AMES FULLER
HENRY DECATUR GORDON
H. K. HATHAWAY
EDWARD J. KEARNEY
WILLIAM LODGE
FRED. E. ROGERS
N. E. ZUSI

### *RAILROADS*

EDWIN BRITTON KATTE, *Chairman*
GEO. M. BASFORD
W. G. BESLER
A. H. EHLE
THEO. N. ELY
W. F. M. GOSS
A. L. HUMPHREYS
WILLIAM FREDERIC KIESEL, JR.
WM. BANCROFT POTTER
NORMAN WILSON STORER
HENRY H. VAUGHAN
ROY V. WRIGHT

### *TEXTILES*

CHARLES T. PLUNKETT, *Chairman*
EDWARD W. THOMAS, *Secretary*
DANIEL MOORE BATES
JOHN ECCLES
E. D. FRANCE
EDWIN FARNHAM GREENE
FRANKLIN WARREN HOBBS
C. R. MAKEPEACE
CHARLES HENRY MANNING
H. F. MANSFIELD

## SPECIAL COMMITTEES

### *CHANGES IN THE PATENT LAWS OF THE UNITED STATES*

WM. HUTTON BLAUVELT
CARL THOMAS
BENJAMIN F. WOOD
EDWARD WESTON
W. E. WINSHIP

### *CODE OF ETHICS*

CHARLES WHITING BAKER, *Chairman*
CHARLES T. MAIN
E. D. MEIER
SPENCER MILLER
CHARLES R. RICHARDS

### *COMMITTEE ON BUREAU OF INFORMATION RESPECTING ENGINEERING STANDARDS IN ALL COUNTRIES*

HENRY HESS, *Chairman*
JOHN H. BARR
CHARLES JACKSON DAVIDSON
CHARLES DAY
CARL SCHWARTZ

### *COMMITTEE TO FORMULATE STANDARD SPECIFICATIONS FOR THE CONSTRUCTION OF STEAM BOILERS AND OTHER PRESSURE VESSELS AND FOR CARE OF SAME IN SERVICE*

JOHN A. STEVENS, *Chairman*
WM. H. BOEHM
ROLLA C. CARPENTER
RICHARD HAMMOND
CHARLES L. HUSTON
HERMAN CHAS. MEINHOLTZ
EDWARD F. MILLER

### *COMMITTEE ON INTERNATIONAL ENGINEERING CONGRESS,* 1915

The President
The Secretary
WM. F. DURAND
R. S. MOORE
T. W. RANSOM
C. R. WEYMOUTH

### *COMMITTEE ON NATIONAL MUSEUM*

E. D. MEIER, *Chairman*
G. F. KUNZ
GEORGE MESTA
H. G. REIST
AMBROSE SWASEY

## *CONFERENCE COMMITTEE ON ENGINEERING STANDARDS*

Henry G. Stott, *Chairman*
Albert Frederick Ganz
Carl Schwartz

## *CONSERVATION*

George F. Swain, *Chairman*
Charles Whiting Baker
Luther D. Burlingame
M. L. Holman
Calvin W. Rice

## *FLANGES*

Henry G. Stott, *Chairman*
Albert Carter Ashton
Arthur Raymond Baylis
Walter M. McFarland
William Schwanhausser
John Porterfield Sparrow

## *INCREASE OF MEMBERSHIP*

Irving E. Moultrop, *Chairman*
Fred H. Colvin
Harold V. O. Coes
James V. V. Colwell
Robert M. Dixon
William Robert Dunn
John Parker Ilsley
Edwin Britton Katte
Richard B. Sheridan
Holger Struckmann

## *CHAIRMEN OF SUB-COMMITTEES ON INCREASE OF MEMBERSHIP*

*Atlanta,* Park A. Dallis
*Boston,* Arthur L. Williston
*Buffalo,* Willis H. Carrier
*Chicago,* Fay Woodmansee
*Cincinnati,* John T. Faig
*Michigan,* Herbert W. Alden
*New York,* James A. Kinkead
*Philadelphia,* Thomas C. McBride
*Rochester,* John C. Parker
*St. Louis,* John Hunter
*St. Paul,* Max Toltz
*San Francisco,* Thomas Morrin
*Seattle,* Robert M. Dyer
*Troy,* Albert E. Cluett

## *PIPE THREADS*

Edwin M. Herr, *Chairman*
Wm. J. Baldwin
Geo. M. Bond
Stanley G. Flagg, Jr.
*Paris Representative,* Laurence V. Benet

## *PIPE THREAD GAGES*

E. D. Meier, *Chairman*
George M. Bond
Arthur M. Houser
J. C. Meloon
Charles Andrew Olson
Peter C. Patterson
Henry G. Stott

## *POWER TESTS*

Geo. H. Barrus, *Chairman*
Edward T. Adams
L. P. Breckenridge
D. S. Jacobus
William Kent
Edward F. Miller
Arthur West
Albert Carroll Wood

## *REFRIGERATION*

D. S. Jacobus, *Chairman*
Philip DeC. Ball
Edward F. Miller
Alfred P. Trautwein
Gardner Tufts Voorhees

## *RESEARCH*

### *Sub-Committee on Materials of Electrical Engineering*

Ralph D. Mershon

### *Sub-Committee on Steam*

Richard H. Rice, *Chairman*
Charles James Bacon
Ernst Julius Berg
Wiliam Duane Ennis
Lionel S. Marks
J. F. Max Patitz

### *Sub-Committee on Safety Valves*

Philip G. Darling, *Chairman*
Henry Decatur Gordon
Edward F. Miller
Frederick L. Pryor
Frederic M. Whyte

*SOCIETY HISTORY*

John E. Sweet, *Chairman* — Frederick R. Hutton, *Secretary*
Henry Harrison Suplee

*STANDARD CROSS-SECTION SYMBOLS*

H. de B. Parsons, *Chairman* — Arthur Edwin Norton
Franklin De R. Furman — Bradley Stoughton
John W. Upp

*STANDARDIZATION OF CATALOGUES*

Wm. Kent, *Chairman* — Morris Llewellyn Cooke
James Rowland Bibbins — Walter B. Snow

*STANDARDIZATION OF FILTERS*

George W. Fuller, *Chairman* — Philip N. Engel
James C. Boyd — J. C. Wm. Greth
William Schwanhausser

*STUDENT BRANCHES*

Frederick R. Hutton, *Chairman* — William Kent
George A. Orrok

*TOLERANCES IN SCREW THREAD FITS*

Luther D. Burlingame, *Chairman* — Arthur Ames Fuller
Elwood Burdsall — James Hartness
Frederic G. Coburn — Henry Martin Leland
Fred H. Colvin — Will. Roswell Porter
Frank O. Wells

# OFFICERS OF THE GAS POWER SECTION
## 1913

*CHAIRMAN*
Frederick R. Hutton

*SECRETARY*
George A. Orrok

*EXECUTIVE COMMITTEE*

Chas. Henry Benjamin (1) — H. J. K. Freyn (1)
Wm. Hutton Blauvelt (3) — Fred. R. Low (2)
William Duane Ennis (5) — Irving E. Moultrop (4)
Henry Harrison Suplee (1)

*LITERATURE COMMITTEE*

Rudolph B. Bloemeke, *Chairman* — W. S. Morrison
A. W. H. Griepe — S. I. Oesterreicher
Wm. F. Monaghan — H. G. Wolfe

*MEMBERSHIP COMMITTEE*

Austin F. Stillman, *Chairman* — John Henry Lawrence
Harry Warfield Anderson — J. H. Norris
Harold V. O. Coes — Godfrey M. S. Tait
Fred S. King — Joseph D. Shaw
Carl D. Smith

*COMMITTEE ON MEETINGS*

Wm. T. Magruder, *Chairman* — Arthur Hugh Goldingham
Wm. Hutton Blauvelt — Nisbet Latta
Edwin D. Dreyfus — Helon Brooks MacFarland

# OFFICERS OF AFFILIATED SOCIETY

*PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS*

W. H. Paine, *Vice-President* — A. H. Whatley, *Treasurer*
J. Ansel Brooks, *Secretary*

Note:—Numbers in parentheses indicate number of years the member has yet to serve.

# OFFICERS OF THE STUDENT BRANCHES

| Institution | Date Authorized by Council | Honorary Chairman | Chairman | Corresponding Secretary |
|---|---|---|---|---|
| Armour Inst. of Tech. Chicago, Ill. | Mar. 9, 1909 | G. F. Gebhardt | H. E. Erickson | A. N. Koch |
| Carnegie Inst. of Tech. Pittsburgh, Pa. | Oct. 14, 1913 | W. Trinks | J. B. Patterson | W. M. Sheldon |
| Case School of App. Science Cleveland, Ohio | Feb. 14, 1913 | F. H. Vose | H. C. Mummert | J. B. Whitacre |
| Columbia University..... New York City | Nov. 9, 1909 | Chas. E. Lucke | F. B. Schmidt | H. F. Allen |
| Cornell University....... Ithaca, N. Y. | Dec. 4, 1908 | R. C. Carpenter | J. G. Miller | D. S. Wegg, Jr. |
| Lehigh University....... South Bethlehem, Pa. | June 2, 1911 | P. B. deSchweinitz | W. C. Owen | T. G. Shaffer |
| Leland Stanford Jr. Univ. Palo Alto, Cal. | Mar. 9, 1909 | W. F. Durand | C. P. Campbell | B. M. Green |
| Mass. Inst. of Tech. Boston, Mass. | Nov. 9, 1909 | E. F. Miller | H. W. Treat | L. L. Downing |
| New York University..... New York City | Nov. 9, 1909 | C. E. Houghton | | |
| Ohio State University.... Columbus, Ohio | Jan. 10, 1911 | Wm. T. Magruder | R. M. Wagner | C. L. Newby |
| Pennsylvania State College State College, Pa. | Nov. 9, 1909 | J. P. Jackson | A. L. Foell | R. W. Herzog |
| Polytechnic Inst. of Bklyn. Brooklyn, N. Y. | Mar. 9, 1909 | W. D. Ennis | A. A. Bielek | H. Sand |
| Purdue University....... Lafayette, Ind. | Mar. 9, 1909 | G. A. Young | A. D. Meals | G. F. Lynde |
| Rensselaer Poly. Inst. Troy, N. Y. | Dec. 9, 1910 | A. M. Greene, Jr. | F. E. McMullen | R. L. Manier |
| State University of Iowa.. Iowa City, Iowa | Apr. 11, 1913 | R. S. Wilbur | F. H. Guldner | C. S. Thompson |
| State Univ. of Kentucky. Lexington, Ky. | Jan. 10, 1911 | F. P. Anderson | H. G. Strong | R. T. Thornton |
| Stevens Inst. of Tech. Hoboken, N. J. | Dec. 4, 1908 | A. C. Humphreys | L. F. Bayer | C. H. Colvin |
| Syracuse University...... Syracuse, N. Y. | Dec. 3, 1911 | W. E. Ninde | G. T. Parsons | W. J. Campbell |
| University of Arkansas... Fayetteville, Ark. | Apr. 12, 1910 | B. N. Wilson | M. McGill | C. Bethel |
| University of California... Berkeley, Cal. | Feb. 13, 1912 | Joseph N. LeConte | R. Guillon | P. H. Landon |
| University of Cincinnati.. Cincinnati, Ohio | Nov. 9, 1909 | J. T. Faig | A. O. Hurxthal | E. A. Oster |
| University of Illinois..... Urbana, Ill. | Nov. 9, 1909 | W. F. M. Goss | A. H. Aagaard | H. E. Austin |
| University of Kansas..... Lawrence, Kans. | Mar. 9, 1909 | F. H. Sibley | L. C. Angevine | H. L. Newby |
| University of Maine...... Orono, Me. | Feb. 8, 1910 | Arthur C. Jewett | E. E. Fowler | A. B. Hayes |
| University of Missouri.... Columbia, Mo. | Dec. 7, 1909 | H. Wade Hibbard | R. Runge | L. L. Leach |
| University of Minnesota.. Minneapolis, Minn. | May 12, 1913 | J. J. Flather | A. Buenger | J. A. Colvin |
| University of Nebraska... Lincoln, Neb. | Dec. 7, 1909 | J. D. Hoffman | A. A. Luebs | G. W. Nigh |
| University of Wisconsin... Madison, Wis. | Nov. 9, 1909 | H. J. Thorkelson | M. A. Cook | R. E. Maurer |
| Washington University... St. Louis, Mo. | Mar. 10, 1911 | E. L. Ohle | A. O. Schleiffarth | J. A. Watkins, Jr. |
| Yale University......... New Haven, Conn. | Oct. 11, 1910 | L. P. Breckenridge | L. F. Harder | M. C. Corbett |

# SUMMARY OF MEMBERSHIP

United States

| | | | |
|---|---|---|---|
| Alabama | 22 | Nebraska | 7 |
| Alaska | 2 | Nevada | 2 |
| Arizona | 8 | New Hampshire | 15 |
| Arkansas | 4 | New Jersey | 263 |
| California | 145 | New Mexico | 3 |
| Canal Zone | 14 | New York | 1404 |
| Colorado | 29 | North Carolina | 16 |
| Connecticut | 214 | North Dakota | 3 |
| Delaware | 21 | Ohio | 357 |
| District of Columbia | 44 | Oklahoma | 1 |
| Florida | 3 | Oregon | 11 |
| Georgia | 20 | Pennsylvania | 630 |
| Hawaii | 10 | Philippine Islands | 4 |
| Idaho | 2 | Porto Rico | 5 |
| Illinois | 347 | Rhode Island | 79 |
| Indiana | 87 | South Carolina | 5 |
| Iowa | 22 | South Dakota | 1 |
| Kansas | 17 | Tennessee | 21 |
| Kentucky | 16 | Texas | 33 |
| Louisiana | 32 | Utah | 11 |
| Maine | 20 | Vermont | 21 |
| Maryland | 56 | Virginia | 33 |
| Massachusetts | 507 | Washington | 30 |
| Michigan | 187 | West Virginia | 14 |
| Minnesota | 54 | Wisconsin | 130 |
| Mississippi | 4 | Wyoming | 2 |
| Missouri | 89 | | |
| Montana | 9 | Total | 5086 |

Foreign Countries

| | | | |
|---|---|---|---|
| Africa | 13 | India | 4 |
| Australia | 9 | Italy | 4 |
| Austria | 3 | Japan | 4 |
| Belgium | 6 | Mexico | 14 |
| Canada | 95 | Norway | 1 |
| Central America | 1 | Roumania | 1 |
| Channel Islands | 1 | Russia | 4 |
| China | 3 | Scotland | 2 |
| Cuba | 11 | South America | 18 |
| Dutch East Indies | 1 | Spain | 4 |
| England | 63 | Sweden | 4 |
| Finland | 2 | Switzerland | 3 |
| France | 11 | Turkey | 1 |
| Germany | 19 | West Indies | 1 |
| Holland | 1 | Total | 304 |

# SUMMARY OF MEMBERSHIP

By Residence

December 31, 1913

| | |
|---|---|
| Membership in United States | 5086 |
| Membership in foreign countries | 304 |
| Present address unknown | 4 |
| Total Membership | 5394 |

By Grades

| | |
|---|---|
| Honorary Members | 14 |
| Members | 3700 |
| Associates | 375 |
| Associate-Members | 92 |
| Juniors | 1213 |
| Total Membership | 5394 |

# GAS POWER SECTION
# SUMMARY OF MEMBERSHIP

United States

| | | | |
|---|---|---|---|
| Alabama | 3 | New Hampshire | 1 |
| Arkansas | 1 | New Jersey | 15 |
| California | 10 | New York | 143 |
| Connecticut | 11 | Ohio | 22 |
| Delaware | 1 | Oklahoma | 1 |
| District of Columbia | 3 | Oregon | 2 |
| Georgia | 3 | Pennsylvania | 42 |
| Illinois | 23 | Rhode Island | 6 |
| Indiana | 11 | Texas | 1 |
| Iowa | 2 | Utah | 1 |
| Maine | 3 | Vermont | 1 |
| Maryland | 2 | Virginia | 1 |
| Massachusetts | 25 | Washington | 2 |
| Michigan | 13 | West Virginia | 2 |
| Minnesota | 5 | Wisconsin | 17 |
| Missouri | 7 | | |
| Nebraska | 1 | Total | 381 |

Foreign Countries

| | | | |
|---|---|---|---|
| Belgium | 1 | Japan | 1 |
| Canada | 3 | Mexico | 1 |
| England | 1 | South America | 2 |
| France | 1 | | |
| Germany | 2 | Total | 12 |

By Residence

| | |
|---|---|
| Membership in United States | 381 |
| Membership in foreign countries | 12 |
| Address unknown | 1 |
| Total membership | 394 |

By Grades

| | |
|---|---|
| Members of the Society | 272 |
| Affiliates | 122 |
| Total | 394 |

# STUDENT BRANCHES

| | |
|---|---|
| Armour Institute of Technology | 8 |
| Carnegie Institute of Technology | 32 |
| Case School of Applied Science | 9 |
| Columbia University | 4 |
| Cornell University | 114 |
| Lehigh University | 23 |
| Leland Stanford Jr. University | 21 |
| Massachusetts Institute of Technology | 22 |
| Ohio State University | 7 |
| Pennsylvania State College | 41 |
| Polytechnic Institute of Brooklyn | 8 |
| Purdue University | 7 |
| Rensselaer Polytechnic Institute | 17 |
| State University of Kentucky | 23 |
| Stevens Institute of Technology | 39 |
| Syracuse University | 26 |
| University of Arkansas | 1 |
| University of California | 2 |
| University of Cincinnati | 46 |
| University of Illinois | 26 |
| University of Iowa | 9 |
| University of Kansas | 22 |
| University of Maine | 18 |
| University of Michigan | 58 |
| University of Minnesota | 30 |
| University of Missouri | 23 |
| University of Nebraska | 13 |
| University of Wisconsin | 18 |
| Washington University | 6 |
| Yale University | 25 |
| Total | 698 |

# CONTENTS OF VOLUME XXXV

BALTIMORE, NEW YORK AND MONTHLY MEETINGS

# TRANSACTIONS

OF

# THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

## VOLUME 35—1913

THIS volume comprises the TRANSACTIONS of The American Society of Mechanical Engineers for 1913. It contains mainly the papers and discussion given at the Spring and Annual Meetings for this year, and the group of papers on Steel Car Construction given at the New York meeting in April. There are also brief accounts of the various other local meetings held throughout the year, the annual report of the Council, and such technical reports as have been approved during the year. The reports of the Standing Committees for the year will be found in THE JOURNAL for December, 1913.

## WILLIAM FREEMAN MYRICK GOSS

William Freeman Myrick Goss was elected President of The American Society of Mechanical Engineers at the Annual Meeting, 1912, for the ensuing year. Dr. Goss was born in Barnstable, Mass., October 7, 1859. In the fall of 1877 he entered the then recently established mechanics arts course at the Massachusetts Institute of Technology. Upon completion of the two-year course he was appointed instructor in practical mechanics at Purdue University and at once began there the work of establishing shop laboratories. His first class of five students was given instruction in those lines of work in which he himself had just been trained. From a meager beginning, the outlook broadened rapidly. The equipment was extended, the number of students increased, and new shop laboratories were built In 1883 he became professor of practical mechanics, a title which held for seven years. When he began, there was no college wes the Allegheny Mountains giving systematic courses in shop pr ,

and there was no manual training work in any American high school. He devised courses of practice and developed series of lectures by means of which principles established in the shops could be given wider application. It was a day when school officials were becoming interested in training students in the manual arts, and many distinguished visitors came to see the work of the Purdue laboratories. The great cities of Chicago, Toledo, Louisville and Indianapolis each in turn sought its aid in the establishment of their manual training schools. Certain forms of equipment, especially forges and lathes, originally designed and constructed at Purdue, were made and supplied as complete equipment to school boards in distant localities where new courses of shop practice were being organized. In many such ways the work at Purdue had an important part in ushering in an educational movement of unusual significance.

In 1889, after ten years of this work, Professor Goss was given a leave of absence, and took up his residence in Boston, where he continued from April of that year to a year from the following September. Some work was done at the Massachusetts Institute of Technology, but the greater part of the time was given to self-directed reading and study. In the spring of 1890 he was appointed professor of experimental engineering, and he undertook the active duties of his new office in the fall of that year. Having developed laboratories for elementary training, it was now his task to build laboratories for advanced engineering work. A modest steam engineering laboratory equipped with a compound Corliss engine and a few testing machines was soon in operation. Plans for an extensive engineering building (Purdue's present engineering laboratory) were developed and by the fall of 1891 a portion of the building was constructed. A significant part of the equipment of the new laboratory was a locomotive testing plant designed to serve in an experimental study of locomotive problems in much the same way that an experimental stationary plant could be used in studying the problems of design affecting the performance of stationary engines. This locomotive testing plant was the first of its kind. It was designed in the summer of 1891 while the building which was to contain it was in the process of erection, and was in successful operation in the late fall of the same year. An incident in the process of installing this plant was that of transporting a 100,000-lb. locomotive over the corn-fields and highways which intervened between the nearest track and the laboratory, a distance by the course taken of about a mile and a half. The opportunities which were presented to its possessors at once attracted the attention

of motive power men and of steam engineers. So meager was the information concerning the performance of locomotives that every fragment of truth, however simple or easily obtained, at once became a matter of public interest. The evaporative capacity of the locomotive boiler, its efficiency at difficult rates of power, the power and efficiency of the cylinders, and the effect upon power and efficiency of changes in speed or cut-off were all matters which previous to the introduction of this plant had been but little understood even by those best informed. The behavior of the various parts of the machine as a mechanism and especially the effects produced by the action of the counterbalance in the locomotive drive wheels were all matters concerning which people had theories, but which were first actually developed by the accurate processes of the laboratory at the Purdue plant. Associations of railroad men gave their encouragement and sometimes financial assistance in increasing the output of the plant. The Master Car Builders' Association made the Purdue laboratories its official testing station and was instrumental in installing there a considerable amount of useful and expensive apparatus. The laboratory became an active center for testing not only locomotives, locomotive fuels and locomotive lubricants, but also details of car construction such as wheels, axles, draft-gears, couplers and brake-shoes. The problems awaiting solution were always numerous, and the professor in charge was kept busy outlining the means to be employed in solving them. He was in the beginning responsible not only for the effective use of the railroad equipment to which reference has already been made, but also for the development of laboratories and courses in materials testing, in hydraulics and in the general field of theoretical and applied thermodynamics. He erected buildings, purchased and installed equipment, and in many cases the equipment installed was of his design, and he was required to meet the reasonable expectations of an ever increasing body of students. Hundreds of men who were students at Purdue in the early nineties can testify to the variety of the activities which in their day were in progress in the engineering laboratories.

In 1899 Professor Goss was again granted a year's leave of absence which he spent in travel and study abroad, chiefly in Germany. Upon his return he was appointed Dean of the Schools of Engineering, an office which he continued to hold throughout the remaining eight years of his residence at Purdue. In 1907 he resigned his position at Purdue, after twenty-eight years of service, to take up what seemed to him the larger and more responsible duties of his

present office, namely, that of Dean of the College of Engineering of the University of Illinois.

Professor Goss was given the honorary degree of Master of Arts by Wabash College in 1888 and the honorary degree of Doctor of Engineering by the University of Illinois in 1904. He has been a member of the Society since 1885, and a member of the American Society for Testing Materials and of the Society for the Promotion of Engineering Education from their organization. He was a member of the Executive Committee of the National Advisory Board on Fuels and Structural Materials, and a member of the Jury of Awards in the Transportation Department of the World's Fair of Chicago in 1893. He is a member of the Master Car Builders' Association, of the Master Mechanics' Association, of the Illinois Academy of Science and of the Western Society of Engineers. He is a fellow of the American Society for the Advancement of Science, a member and past-president of the Western Railway Club and was the Chairman of the Advisory Committee organized by the Pennsylvania Railway Company to direct its work in locomotive testing at the Louisiana Purchase Exposition. For many years he was a contributing editor to the Railroad Gazette. His contributions to technical literature have been numerous. Probably his best known work is that which deals with the locomotive. His books on Locomotive Performance and Locomotive Sparks are records of personal researches which are known to most railroad men and to most students of locomotive design. Numerous researches conducted under the patronage of various associations are available only in the proceedings of these associations. Besides these, two noteworthy pieces of work were done under the patronage of the Carnegie Institution of Washington, the results of which are presented in two volumes, one entitled High Steam Pressures in Locomotive Service and the other Superheated Steam in Locomotive Service.

# ANNUAL REPORT OF THE COUNCIL

Covering the presidential year of Dr. W. F. M. Goss the Society has been especially active in the work of the standing and special committees, and the Council takes this opportunity to express its appreciation of the splendid work unselfishly done.

The Committee on Power Tests has reorganized with George H. Barrus as chairman and the committee will within the next year be able to present its completed report of recommendations of Standards for Testing Power Plant Apparatus.

The Committee on Standardization of Flanges has presented recommendations of a new standard to be known as The American Standard, to become effective January 1, 1914. The standard was arrived at after a series of conferences with the Master Steam and Hot Water Fitters Association and the Manufacturers' Committee and is a compromise, consistent with good engineering practice, between the 1912 U. S. Standard heretofore recommended and that recommended the same year by the Manufacturers.

The Committee to Formulate Standard Specifications for the Construction of Steam Boilers and other Pressure Vessels and for Their Care in Service has completed a preliminary report to be sent out to expert boiler men and manufacturers for trial inspection with the idea of getting further criticism before final recommendation and presentation of the report to the Society.

The Sub-Committee on Fire Protection has recommended a National Standard for Hose Couplings, which the Council has ordered published. It was the opinion of the chairman, John R. Freeman, that while this standard is perhaps not ideal, yet it is an expedient that meets a decided need.

The Committee on an International Standard for Pipe Threads has sent to the Paris representative, L. V. Benet, instructions for presentation to the International Commission on Pipe Threads which was initiated by the Société Technique de l'Industrie du Gaz in France. The American Gas Institute has sent identical instructions to its representative.

At the invitation of the American Institute of Electrical Engineers a committee was appointed to coöperate with its Standards Committee

and report concerning the use of the Myriawatt as a unit. This report was presented and made part of the proceedings of the Spring Meeting.

At the request of a number of members and manufacturers a committee was appointed and has presented its report on the fixing of manufacturing limits in flanges and fittings threaded according to the Briggs Standard. Similarly a committee was appointed on a standard of rating the capacity of mechanical filters.

Wilfred Lewis, chairman of the Committee on Involute Gears, has presented a majority report of his committee and the committee has been discharged with the thanks of the Council.

A report is before the Council recommending the appointment of a committee to act as a clearing house for all matters of standardization. The special committee to outline this work consists of Henry Hess, chairman, J. H. Barr, Charles Day, C. J. Davidson, and Carl Schwartz. This matter has been undertaken in response to the many inquiries which are received by the Society and in recognition of the fact that standardization is essential to engineering. The bureau does not propose to do any work of standardization for the various engineering organizations of the world, but aims solely to bring together information obtained from all bodies interested in standardization of similar lines and promptly to inform each body of the proceedings of every other organization engaged in similar work.

At the Spring Meeting the Committee on a Code of Ethics presented for discussion its draft of a proposed Code of Ethics. This code has been mailed to the membership for letter ballot giving each member the opportunity to vote for or against it as a whole or in part, or advising amendments.

At various functions during the year the Society has been represented by the appointment of Honorary Vice-Presidents as follows: Annual Convention of the Southwestern Electrical and Gas Association, Galveston, Texas, W. B. Tuttle; Third International Refrigeration Congress, reception to delegates in New York, H. G. Stott, J. W. Lieb, Jr., Calvin W. Rice; on committee in charge of first section devoted to liquid gases and units, William Kent, president, D. S. Jacobus, vice-president, Calvin W. Rice, secretary, Louis Block, Charles E. Lucke, E. F. Miller, S. W. Stratton; Third International Drainage Congress, John Hunter; National Society for the Promotion of Industrial Education, convention in Chicago, F. A. Geier, Charles R. Richards, M. E. Cooley, John C. Bird; American Association for the Advancement of Science, members of council, Alex. C. Humphreys, W. B. Jackson; Chattanooga Chamber of Commerce, at opening and

dedication of Chattanooga and Tennessee River Power Company's hydroelectric development, Newell Sanders.

The Society has lost by death during the past year the following Honorary Members: John Fritz, Carl P. deLaval, Sir Wm. H. White, V. Dwelshauvers-Dery, Rudolph Diesel, Sir Wm. Arrol.

Honorary Membership has been conferred upon Charles H. Manning.

The report in changes in membership in the following table covers the fiscal year October 1, 1912, to September 30, 1913, in distinction from the administration year which the rest of the report concerns.

MEMBERSHIP FOR FISCAL YEAR (OCT. 1, 1912–SEPT. 30, 1913)

| Grade | Oct. 1, 1912 | Losses | | | | Additions | | | Net Decrease | Net Increase | Oct. 1, 1913 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Transfer | Resignations | Lapses | Deaths | Transfer | Election | Reinstatement | | | |
| Honorary | 18 | .. | .. | .. | 5 | 1 | 1 | .. | 3 | .. | 15 |
| Member | 2961 | 1 | 17 | 19 | 30 | 81 | 468 | 4 | .. | 486 | 3447 |
| Associate | 357 | 18 | 8 | 10 | 6 | 1 | 41 | 3 | .. | 3 | 360 |
| Junior | 1006 | 64 | 12 | 29 | 3 | .. | 263 | 3 | .. | 158 | 1164 |
| Total | 4342 | 83 | 37 | 58 | 44 | 83 | 773 | 10 | .. | 644 | 4986 |
| Affiliates Gas Power Section | 126 | .. | .. | .. | .. | .. | .. | .. | 2 | .. | 124 |
| Affiliates Student Section | 722 | .. | .. | .. | .. | .. | .. | .. | 73 | .. | 649 |

The Increase of Membership Committee, with the ready and splendid response and coöperation of the membership at large, is doing most effective work on the recommendation of the committee. The Council has appointed sub-committees in different centers under the chairmenship of the following: Park A. Dallis, Atlanta; A. L. Williston, Boston; W. H. Carrier, Buffalo; Fay Woodmansee, Chicago; J. T. Faig, Cincinnati; R. B. Sheridan, Cleveland; H. W. Alden, Michigan; J. A. Kinkead, New York; T. C. McBride, Philadelphia; John Hunter, St. Louis; Max Toltz, St. Paul; Thos. Morrin, San Francisco; R. M. Dyer, Seattle; A. E. Cluett, Troy.

Student branches have been added at Case School of Applied Science, State University of Iowa, University of Minnesota, and for

students taking the regular technical course in applied science at Carnegie Technical School, making a total of 30 branches with 650 student affiliates enrolled.

As the committee and representative of the Society in San Francisco in the matter of the International Engineering Congress in 1915, W. F. Durand, T. W. Ransom, C. R. Weymouth, R. S. Moore, the President and the Secretary of the Society, ex-officio, have been appointed and much work of organization has been started.

There is given elsewhere the record of the official visit of this Society to Germany and the joint meeting with the Verein deutscher Ingenieure, June 21 to July 7, 1913. The trip through Germany was primarily to observe the engineering and industrial work of the nation and every opportunity was afforded for the inspection of the leading plants of every city. At Leipzig were the two professional sessions of the Verein in which The American Society of Mechanical Engineers participated.

Important amendments have been made to the Constitution and By-Laws, the principal changes being occasioned by the additional grade of Associate-Member and the change in the method of electing members. An Associate-Member is defined as follows:

C-11 An Associate-Member shall be an Engineer or Teacher of Applied Science of twenty-five years of age or over. He must show by his experience or by his duties that he is competent to execute work in his profession.

The change in the method of balloting for the members is indicated in the following:

C-15 All applications for membership to the grades of Member, Associate, Associate-Member or Junior shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council shall, unless objection is made by the applicant, be submitted to all the members of the Council for election, by means of a letter-ballot.

C-16 Any person desiring to change his grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.

C-17 Election to membership shall be by sealed letter-ballot of the Council as the By-Laws shall provide. Fifteen affirmative votes shall be required for the election of a candidate to membership in any grade. One negative vote shall defeat an election to Honorary Membership. Two negative votes shall defeat an election to any other grade.

The Council favors even more strict requirements in the grades of Member and Associate-Member and at the Spring Meeting presented the following proposed amendment:

C-9 A Member shall be an Engineer or Teacher of Applied Science of thirty-two years of age, or over, and shall have been in the active practice of his profession for at least ten years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a Professor of Engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice.

C-11 An Associate-Member shall be a professional engineer not less than twenty-seven years of age, who shall have been in the active practice of his profession for at least six years, and who shall have had responsible charge of work as principal or assistant for at least one year. Graduation from a school of engineering of recognized reputation shall be considered as equivalent to two years' active practice.

Local sectional meetings, additional to previous years, are now being held in Chicago, Atlanta, Milwaukee and St. Paul-Minneapolis, making a total of 11 cities in which there is an organization with periodical meetings held.

On the recommendation of the Committee on Meetings an additional sub-committee has been appointed on Depreciation and Obsolescence, to determine as far as possible the proper annual charges to operating expense for depreciation—including physical decay, obsolescence and inadequacy—in the various industries in which the members are interested.

The Society's Committee on Conservation received the approval of the Council for presentation of its protest to the Congressional committee, covering the transfer of the control of the national forests to the individual states. This was one of the subjects brought up for consideration at the Fifth National Conservation Congress in Washington, November 18, 19 and 20, to which the Council appointed as its representatives the members of the Conservation Committee and General William H. Bixby, Admiral H. I. Cone, Dr. Joseph A. Holmes. The congress was this year devoted mainly to Forestry and Water Power. The work of Conservation in general is one in which The American Society of Mechanical Engineers may well take a thoughtful interest and part.

The Baltimore Engineers Club has been added to the societies listed on members introduction cards with whom we exchange library and house courtesies.

At the beginning of the year the Council referred to the Publication Committee the study of the various publications of the Society with a view to increasing their value to the membership and decreas-

ing the cost. The committee has responded with a plan which has the unanimous endorsement of the Council. This involves the issuing of The Journal in the 9 by 12 size now so generally used by technical journals and the handling of the advance papers and the matter in The Journal as follows:

*a* Papers to be printed in pamphlet form (9 x 6 size) complete in advance of a meeting, but not published in The Journal prior to the meeting.

*b* Abstracts (of perhaps 500 words each) of all the papers for a given meeting to be grouped and published in one number of The Journal at least one month in advance of the meeting.

*c* Announcement to be made each time by circular, return postal or in The Journal, that copies of complete papers on any subject will be sent free to any member asking for them.

*d* That papers and discussion be published together in issues of The Journal following the meeting at which they are presented.

By this procedure there will appear in The Journal at an earlier date than is possible in the annual volume of Transactions the revised papers and discussion given at meetings, and when published in this way The Journal will contain the Transactions of the Society in complete form and will be worthy of binding and preservation by the membership.

No. 1387

# MEETINGS JANUARY—JUNE

## MEETINGS PREVIOUS TO THE SPRING MEETING

### NEW YORK, JANUARY 14

Meeting devoted to discussion of work of the New York Committee. A full account appeared in The Journal for February, 1913.

### BOSTON, JANUARY 22

Joint meeting under the auspices of the Boston Society of Civil Engineers. Hydraulics in City Building: Paper presented by William H. Lewis, president Lewis Wiley Hydraulic Company, Seattle, Wash., and Portland, Ore. An account appeared in The Journal for February, 1913.

### CHICAGO, FEBRUARY 6

Internal-Combustion Engines: Discussion by E. T. Adams, giving the author's experiences and discussing the possibilities of building large gas engine units; by Prof. Chas. R. Richards; by J. C. Miller, consulting engineer, Chicago; and by Nisbet Latta, followed by general discussion. An account of the meeting appeared in The Journal for March, 1913.

### ST. LOUIS, FEBRUARY 5

Paper: Lubricating Value of Cup Greases, A. L. Westcott, University of Missouri, describing tests of cup greases.

An abstract of this paper appears in The Journal for July, 1913.

### PHILADELPHIA, FEBRUARY 8

Paper: Overhead Expense Distribution, by Royal R. Keely, discussing the various methods employed in computing this important item and their relative merits.

An abstract with discussion appears in The Journal for June, 1913.

NEW YORK, FEBRUARY 11

Paper: Port Facilities for Ships and Cargos in the United States, by Wm. T. Donnelly, describing the present situation in various ports.

An abstract of this paper and discussion appears in The Journal for June, 1913.

BOSTON, FEBRUARY 25

Papers: Some Thermal Properties of Concrete, C. L. Norton; Experience with Concrete in Fires, G. E. Fisher, engineer, Arkwright Mutual Fire Insurance Company.

Abstract of the papers and discussion appears in The Journal for June, 1913.

NEW YORK, MARCH 13

Joint Meeting with the Illuminating Engineering Society, the New York Association for the Blind, and the American Museum of Safety. Papers: Illumination and Eyestrain, Dr. Ellice M. Alger; Industrial Lighting Problem from the Standpoint of the Illuminating Engineer, Ward Harrison; the Problem from the Standpoint of the Mechanical Engineer, A. C. Jackson. Dr. W. H. Tolman of the American Museum of Safety showed a number of slides showing the importance of lighting in the problem of safety.

A more complete account appears in The Journal for April, 1913.

MINNEAPOLIS, MARCH 18

Organization meeting and dinner. Addresses by Calvin W. Rice, Secretary, Prof. J. J. Flather, J. L. Record, Oliver Crosby, C. L. Pillsbury, and Max Toltz. A more complete account appears in The Journal for April, 1913.

ST. LOUIS, MARCH 19

Address: Engineering and Common Sense, William Kent.

CINCINNATI, MARCH 20

Address: Engineering and Common Sense, William Kent.

BOSTON, MARCH 25

Papers: Some of the Problems Encountered in the Design, Construction and Equipment of the Modern Cotton Mill, Frank

W. Reynolds; Modern Methods of Lighting in Cotton Mills, Albert L. Pearson; Air Conditioning for Textile Mills, Fred. W. Parks, president, G. M. Parks Company, Fitchburg, Mass.

An abstract of these papers with discussion appears in The Journal for June, 1913.

NEW YORK, APRIL 4

Joint meeting with the American Institute of Mining Engineers and the American Institute of Electrical Engineers. Demonstration of kinetophone and latest developments in the moving picture and phonograph fields by Miller Reese Hutchinson, through the courtesy of Thomas A. Edison, Honorary Member. A more complete account of the meeting appears in The Journal for May, 1913.

NEW YORK, APRIL 8

Meeting under the auspices of the Sub-Committee on Railroads and the New York Committee on Meetings. Subject: Symposium on Steel Passenger Car Design. Published in this issue of Transactions.

PROVIDENCE, APRIL 10

Joint dinner with the Providence Association of Mechanical Engineers, with addresses by Prof. T. M. Phetteplace, Calvin W. Rice, Secretary, Prof. W. H. Kenerson, Prof. C. F. Scott. A more complete account appears in The Journal for May, 1913.

NEW HAVEN, APRIL 16

Quarterly meeting with afternoon and evening sessions. Papers: General Types of Heating Systems, by Allen C. Staley of Sheffield Scientific School; Heating, Ventilating and Humidifying of the Cheney Brothers' Silk Mills, by G. H. Miller; Applications of Lighting to Factories, by C. E. Clewell of Sheffield Scientific School; Latest Improvements in Construction of Gas Lamps, by T. J. Lytle, Welsbach Company, Philadelphia.

A more extended account appears in The Journal for May, 1913.

BOSTON, APRIL 25

Joint meeting under the auspices of the American Institute of Electrical Engineers. Paper: Delivery and Handling of Freight at the Boston Freight Terminals, by Harold Pender, H. F. Thompson

and C. P. Eldred, describing prevailing conditions and possible improvements.

CHICAGO, MAY 7

Dinner, with speeches by Robert W. Hunt, Philetus W. Gates, and George M. Brill. A more complete account appears in The Journal for June, 1913.

SAN FRANCISCO, MAY 15

Paper: Progress of Buildings being Erected for Panama-Pacific International Exposition, by G. L. Bayley.

## THE SPRING MEETING

The meeting at Baltimore was in every way a successful gathering. The Society and its friends were the guests of the Engineers' Club of Baltimore, with whom the local members of the Society and the officers at Annapolis coöperated to the fullest extent; and the work of the local committees under the direction of the Executive Committee of the Engineers' Club was planned and carried out with such nicety of detail as greatly to enhance the enjoyment of all in attendance.

The registration was small as had been anticipated, in view of the approaching German Meeting, there being 142 members present and 185 guests. There have seldom been professional sessions at any of the Society's meetings, however, at which there was greater sustained interest or more effective discussion. The program follows:

### PROGRAM

*Tuesday Afternoon, May 20*

Registration of members and guests at headquarters, Hotel Belvedere.

*Tuesday Evening*

Membership reunion and informal reception.

*Wednesday Morning, May 21*

BUSINESS MEETING

Reports of tellers of election of members. Announcement of ballot on amendments to the Constitution relating to membership grades; new business. Reports of Special Committees on Myriawatt, Involute Gears, Code of Ethics.

SIMULTANEOUS SESSIONS FOLLOWING BUSINESS MEETING

PROFESSIONAL SESSION

TEST OF A HYDRAULIC BUFFER, Carl Schwartz.

Discussed by F. H. Clark, A. E. Johnson, H. A. Jensenius, Philander Betts.

THE PRESENT CONDITION OF THE PATENT LAW, Edwin J. Prindle.

Discussed by J. N. McGill.

Shading in Mechanical Drawing, Theodore W. Johnson.

Discussed by S. A. Moss, W. P. Hawley, H. D. Hess, L. S. Burbank, L. E. Osborne, L. D. Burlingame, F. W. Ives, J. S. Reid, J. G. Matthews. Published in The Journal, April and August, 1913.

Cost of Upkeep of Horse-Drawn Vehicles against Electric Vehicles, W. R. Metz.

Discussed by H. H. Smith, A. M. Pearson, L. H. Flanders, W. P. Kennedy, John Younger, E. R. Gurney, Harrington Emerson, C. W. Baker.

GAS POWER SESSION

Present Operation of Gas Engines using Blast-Furnace Gas as Fuel, Charles C. Sampson.

Discussed by F. H. Wagner.

*Wednesday Afternoon*

Demonstration of the high-pressure fire system at City Hall plaza and inspection of pumping station, followed by a sail about the harbor to inspect the water front, shipping facilities, and other features of interest.

*Wednesday Evening*

Lecture illustrated by lantern views: Around the World in Eighty Minutes, by Hon. O. P. Austin, Secretary, National Geographic Society.

*Thursday Morning, May 22*

FIRE PROTECTION SESSION

The Baltimore High-Pressure Fire Service, James B. Scott.

National Standard Hose Couplings and Hydrant Fittings for Public Fire Service, F. M. Griswold.

Published in The Journal March 1913.

Debarment of City Conflagrations, Albert Blauvelt.

Allowable Height and Area in Factory Buildings, Ira H. Woolson.

The Protection of Main Belt Drives with Fire Retardant Partitions, C. H. Smith.

The Life Hazard in Crowded Buildings due to Inadequate Exits, H. F. J. Porter.

Discussed by W. H. Kenerson, Henry Hess, G. I. Rockwood, Harrington Emerson, F. B. Gilbreth.

*Thursday Afternoon*

Inspection of sewage pumping plant, Jones Falls conduits, and trip by trolley to sewage disposal plant at Back River.

Automobile trips for ladies, about the city and suburbs, with tea served at the Country Club.

*Thursday Evening*

Reception and dance tendered by the Engineers' Club of Baltimore.

*Friday, May 23*

All-day excursion to Annapolis and to the U. S. Naval Academy with a reception by Governor Goldsborough at the State House. After the reception the party proceeded to the Assembly Chamber where Admiral H. I. Cone, engineer-in-chief of the Bureau of Steam Engineering, U. S. N., delivered an address upon the United States Experimental Station at Annapolis.

Luncheon at Carvel Hall and trip to Naval Experiment Station and aviation camp, with demonstration flights.

# SYMPOSIUM ON STEEL PASSENGER CAR DESIGN

No. 1388 *a*

## INTRODUCTION

By H. H. Vaughan, Montreal, Canada

Member of the Society

The advent of the steel passenger car has brought with it many new problems and an opportunity for more diverse opinions than any other change that has taken place in car equipment. The construction of the wooden passenger car developed along fairly uniform lines. The varieties of framing were few and the differences unimportant, while the introduction of steel platforms, wide and narrow vestibules, reinforced end and sill construction and similar improvements occurred gradually, and with practically similar designs on all railroads. The change from wood to steel in freight car construction resulted in the abandonment of designs that had almost become standardized and the introduction of many new types, but in this case the principal problem, other than that of obtaining satisfactory designs, has been the extent to which it was advisable to use composite or all-steel construction.

2 In the case of the passenger car, the types to be employed will probably not be changed by the substitution of steel for wood. The increase in capacity that has taken place in freight equipment cannot be duplicated in passenger cars, and there appears to be no tendency at present toward any increase in length or carrying capacity. The questions that now confront us relate rather to the design and construction of cars of the present type and of the materials that may be advantageously employed in place of the wood which has been used for so long. They are complicated by the necessity of providing for greater safety for the passengers than was secured in the wooden car, with an equal degree of comfort and the difficulty of anticipating the behavior of this new equipment in the case of accident. Certain difficulties such as the best systems for heating, lighting and ventila-

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

tion, are common to both steel and wood construction, and improvements in these matters pertain to general progress rather than the use of steel construction. The following list, while probably incomplete, outlines in a brief way the important variations that must be considered in deciding on the preferable construction of steel passenger equipment:

| | | |
|---|---|---|
| Framing | Steel underframe | |
| | All-steel frame | Center girder<br>Side girder |
| Outside finish | Plated<br>Sheathed | |
| Roof construction | Clear story<br>Circular | |
| Inside finish | Steel<br>Wood | |
| End construction | Design and strength | |
| Floor | Design and material | |
| Insulation | Material | |

No doubt questions of equal importance have been omitted, and in many cases those mentioned require careful consideration with regard to degree, as for instance, the strength of the framing or the thickness of the insulation. The list illustrates, however, the diversity of possible solutions of the preferable steel passenger car, and the following personal opinions are presented for the purpose of opening the discussion:

3 The steel underframe does not appear to be a satisfactory or permanent development. There is but little saving either in weight or cost over the all-steel construction, and it is difficult to see how the same strength in case of accident can be obtained. Experience will show whether the wood superstructure can be secured in such a way as to prevent working as the car gets old, but as it cannot be arranged to carry any weight this appears questionable. It can hardly be regarded except as an intermediate step between all-wood and all-steel construction.

4 In all-steel construction the side-girder car presents advantages, but as in freight construction, both types will probably persist. The side-girder construction obtains greater strength on the side framing without superfluous weight, and it is possible that greater framing strength may prove necessary. With equal strength of side framing the side-girder car may be made lighter than the center-girder type, and the weight of steel passenger cars is one of the most serious problems to be faced by any railroad not having a level line. Ameri-

can passenger equipment was already excessively heavy per passenger carried with wood construction, and the use of steel has increased this weight from 10 per cent to 20 per cent, which is a most serious matter. Apparently side-girder cars as so far constructed have a decided advantage over the center-girder type in their light weight and greater strength in case of accident tending to crush in the side of the car. This will probably lead to the use of this type on roads on which weight is of importance.

5 In spite of the many advantages of the sheathed car in case of construction and maintenance, it appears that the cost and weight of the additional metal will prevent its extensive use. This question is chiefly one of appearance and convenience, and is of minor importance.

6 The circular roof has been extensively introduced on steel passenger cars on account of its lightness and simplicity of construction. It has the objection that deck sash ventilation cannot be employed. The Pullman Company while using the clear-story roof have, however, discontinued the use of deck sash ventilation, so that evidently in their opinion this objection is not important. The deck sash is, however, of value in a standing car, and when properly screened is certainly advisable in hot weather, especially when the road is dusty. The Canadian Pacific Railway have compromised on this question and are using a roof of approximately circular form with deck sash. The strength and simplicity of the circular roof is retained with the ventilating qualities of the clear story type.

7 The preferable material for inside finish is a matter for future decision. With the ample protection afforded by a steel car against accident, there does not appear to be any objection to wood inside finish on the ground of safety. It is more ornamental than steel and a better insulator. Probably on no question in passenger car design is opinion so divided amongst both railroad and car builders. There is today very little difference in cost, and it certainly appears probable that in the future the tendency will be to adopt steel interior finish if not entirely, at any rate to a great extent.

8 The construction of the ends of the cars has received considerable attention, and the strength now usually employed is enormously greater than anything attempted in wood construction. Several excellent designs have been devised, which will probably be referred to in another paper.

9 The floor construction in steel cars is entirely different from that in wooden cars, and is usually of metal covered with a flexible

cement. In constructing a sample car for the Canadian Pacific Railway the writer used in addition an underfloor covered with insulating material, and covered the cement with ½ in. of cork. This car was also exceptionally well insulated at the sides, 2 in. of cork being used next the outside plating. Tests during the past winter have shown that this car is actually warmer than the ordinary wooden car, the same amount of heating surface being used in both types. The floor was tested by taking the temperature of water standing in cans on the floor, there being no practical difference between the results in the wood and steel cars. The question of insulation is an important one, both in hot and cold weather, and while other insulation might no doubt be equally effective, it is interesting to be able to advise that with proper insulation there is no question of the steel car being satisfactory.

## No. 1388 *b*

# PROBLEMS OF STEEL PASSENGER CAR DESIGN

By W. F. Kiesel, Jr., Altoona, Pa.

Member of the Society

Whenever it becomes necessary to adopt a policy representing a complete departure from existing policies involving a new theoretical structure from foundation up, many problems, some entirely new, have to be solved. The increasing cost of lumber, the desire for longer and stronger cars, and other considerations indicated the desirability of making a determined effort to develop a satisfactory steel passenger car. The object of this paper is to review a few of the problems encountered, beginning with:

2 *First: Can we afford it, and what will it cost, compared with wooden cars?* Tentative designs were prepared and carefully analyzed by a committee of representatives of carbuilders and railroads. The summary of their report was that at first steel passenger cars would cost approximately 20 per cent more per passenger than wooden cars of the best existing types, but that the steel cars would probably cost much less to maintain. They also reported that on account of the increasing cost of good lumber, and the probable decreasing cost of manufacturing steel cars, not many years would elapse before the cost of steel cars would be no more than, if as much as, wooden cars. Those who have been in close touch with the development of the steel-car industry know that at the present time steel cars cost no more than equivalent wood cars.

3 *Second: Shall the cars be all steel, or steel frame with wood lining?* Differences of opinion still exist on this point. Both types of car have been built, and each has strong advocates.

4 In the all-steel car the steel lining can be securely riveted to the framing and adds somewhat to the strength of the complete structure, but as steel is a good conductor it carries away the heat of a body coming in contact with it, and, therefore, will always feel cold, even when the temperature in the car is sufficiently high. Satisfactory results have been realized from the use of a double steel lining

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

between seats, forming a hot-air duct, extending from the heater pipes to the window sill, with outlet through small holes in the lining proper, located immediately below the window sill in the lining proper.

5 Wood lining requires considerable wood furring, and adds weight to the car without adding to the strength. As the steel frame of a long passenger car may vary as much as ½ in. between extremes of temperature, it is necessary to make allowance in the construction of the wood lining for this variation in length. As a car with metal lining riveted to the framing has the advantage in strength, weight, and cost, it will gain in favor; in fact, it would be at present universally preferred if all railroad shops had practical experience with steel lining, and the necessary proficiency and machinery for its manufacture.

6 *Third: Insulation.* Three general principles have been used for car insulation: (*a*) Wood lining; (*b*) by placing insulating material on the outside of steel lining; (*c*) by placing insulating material on the outside of the steel lining, and on the inside of the steel sheathing.

7 Experiments have been made also with other methods, such as completely filling the space between sheathing and lining with block magnesia and magnesia cement. The problem that presents itself is: Given a car body with a comparatively smooth exterior surface protected by several coats of paint, double walls, painted on both sides—if of steel, isolated air spaces, rather large in volume, between the walls, an inside cubic volume in which the air must be continually renewed, and a window surface of about one-third of the area of the side walls. When single windows are used the air close to the windows is cold in winter, and warm in summer. Double windows improve the situation materially.

8 Experiments made to determine the difference between a wooden and a steel coach, with doors and windows closed, standing on a siding exposed to the sun in hot, summer weather, showed a difference of one to two degrees in favor of the wooden coach. One day's readings showed an average of one degree difference in temperature in favor of the steel coach, which had insulation only on the outside of the lining. The results of several years' experience indicate that the lining must be insulated throughout, and, if the spaces between lining and sheathing are properly isolated, little is gained by insulating the sheathing, and more will be gained by the use of double windows. Furthermore, the heat lost in cold weather by conduction through and radiation from the walls, in cars with insula-

tion on the lining alone, is negligible when compared with the heat carried off by adequate ventilation.

9 *Fourth: Protection and safety of passengers.* This problem involves providing adequate strength for carrying the load, also to prevent collapse or crushing in wrecks, and efficient brakes.

10 The laws governing load-carrying strength are well known, but this cannot be said of the laws governing wrecks. Each wreck forms a separate study, and we seldom find two that can be placed in the same class. The study of wrecks, which, unfortunately, do occur, shows that the car underframe must be reasonably strong to resist end strains, that the ends of the superstructure must be reinforced with strong vertical members, and that the car must not collapse when rolled down an embankment. The gradual elimination of crossings at grade has materially decreased the danger of strains directed against the sides of the car.

11 Early experience with steel freight cars showed clearly that the men handling cars in yards believed that all cars built of steel could withstand much rougher handling than wooden cars. Although the resultant damage to both kinds of freight cars had its disadvantages, it developed a better knowledge of the relative value of steel and wood in car construction, led the designer to abandon the basis of ultimate strength of the material, and to substitute the basis of elastic limit, and finally to select a ratio of 4 to 1 as the relation of the elastic limit of steel as used in cars to that of good timber.

12 That not all designers of steel passenger cars had the advantage of this knowledge, or profited by this experience, is evidenced by some of the car designs which have been illustrated in the technical papers in the past years and which proved fundamentally defective.

13 Selecting from the last generation of wooden cars one used in heavy trunk line service, with four 5-in. by 9-in. wooden sills bunched together near the center, and so located as to be nearly uniformly affected by the end strains, steel platforms with draft gear securely attached, and the remainder of the car to correspond, the analysis of its end-shock resisting capacity leads to the consideration of the elasticity of the material, the transverse bracing preventing buckling, the concentration of strength near the longitudinal center line of car, and the reinforcement at the platforms.

14 The wooden car, therefore, meets many of the requirements enumerated before. A corresponding steel car should have a center sill area of 45 sq. in. braced against buckling, a strong and efficient draft gear as a substitute for the elasticity of the wood, and a ratio of 0.04

for stress to end force, the calculations to include consideration of lever arm of force below neutral axis of the center sills. For lighter service a steel car with center sill area of 32 sq. in. and a ratio of 0.05 for stress to end force may be considered as a substitute for a wooden car with four 4-in. by 8-in. sills bunched near the center of the car. The use of steel permits a distribution of material to better advantage than is possible with wood. The box girder center construction is continually gaining in popularity, the strong vertical members at car ends, to prevent one car overriding and penetrating the superstructure of another car, are now considered a necessity, and a superstructure, including roof sufficiently strong to bear the car when turned upside down without collapsing, is very desirable.

15 To avoid making this paper too long other interesting problems will be omitted, but the truck problem deserves brief consideration. There are four-wheel and six-wheel trucks. They have 4¼-in. by 8-in., 5-in. by 9-in., 5½-in. by 9-in. and 5½-in. by 10-in. journals.

16 The impression that cars with six-wheel trucks necessarily have better riding qualities than those with four-wheel trucks has proved to be incorrect. The substitution of four-wheel trucks for six-wheel trucks saves about 18,000 lb. per car. Increased journal bearing surface obtained by an increase of diameter of journal only is of little or no benefit in preventing hot boxes, because the periphery velocity increases in the ratio of the diameters. The weight per journal should not exceed 1500 lb. per in. length. A long spring base, low-lying center plate, and anchoring the dead levers to the car body instead of to the truck frame promote smooth action and easy riding at all times. The equalizing springs should, therefore, be placed as near to the journal boxes as possible, or directly over the boxes, and the bolster springs should be on or near the center line of truck sides. If the dead levers of the truck brake are anchored to the car body, the truck frames have no tendency to tip up when the brakes are applied, and the jarring effect is entirely eliminated. A special axle with 5½-in. by 11-in. journal for passenger cars would be of material benefit, would permit using four-wheel trucks under all coaches and 60-ft. baggage cars, and longer cars with six-wheel trucks would have sufficient margin for the excessive loads sometimes encountered and the danger of hot boxes would be avoided.

No. 1388 c

# UNDERFRAMES FOR STEEL PASSENGER CARS

By John McE. Ames, New York

Member of the Society

This paper will be confined to underframes of steel passenger cars for through service, or those at least 70 ft. long, and will not attempt to discuss those of suburban or individual service, whose underframes are not subjected to the same severe service strains.

2 The underframe is called upon to perform several functions. Not only must it sustain the weight of the superstructure and load, but withstand impact, oscillation and pulling strains without distortion. Were it not for these conditions the underframe might be considered as a bridge resting upon the center plates and side bearings as piers. Were we to design to meet only the carrying requirements the problem would not be difficult, but the design must also be commercial, not over heavy and in addition sufficiently strong to resist impact; commercial in that plates and shapes employed are such as may readily be secured from the steel mills, and not so heavy as to bring undue work upon either the hauling locomotive, rails, frogs, bridges, etc.

3 The natural division of such designs is:

*a* Underframes designed to carry equally on all sills

*b* Underframes designed to carry on center sills only

*c* Underframes designed to carry on sides only

*c* Underframes designed to carry on sides and center sills

4 Each of these types has its partisans and each type is in successful operation today. The first is the type used abroad almost universally and at home for repairs under wooden cars, the bodies of which are too good to destroy but need better underframing. With most of the foreign cars the body rests upon and is bolted to the underframe from which it may readily be removed. The buffing and draft conditions differ from ours in that the buff is taken through

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

the side sills by the use of separate side buffers, and the draft through the center sills thus permitting a distribution of metal in each sill member that may produce uniform stress.

5 An example of the first type designed for a wooden superstructure, consists of four deep sills of what is known as the "fishbelly" type (Fig. 1). These center sills are composed of 5/16-in. plates, 30 in. deep at the center with 3 in. by 3 in. by ⅜ in. angles riveted along the top and bottom edges; the plates reduced to a depth

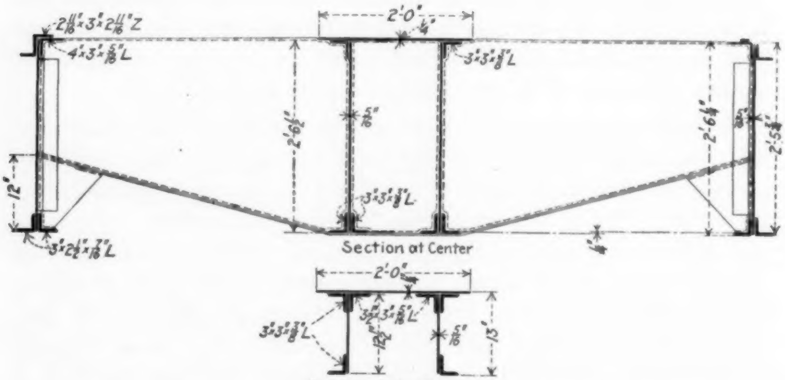



FIG. 1 TYPE *a*: WEIGHT OF CAR CARRIED EQUALLY ON CENTER AND SIDE SILLS

of 12¾ in. over the bolster. The center sills have a square inch section of 37 at the center, just as the side sills, and 26 at the draw gear. One disadvantage in these long plate sills is that when punching the line of holes along the edges the plate becomes distorted and wavy. It is then difficult to rivet the angles in place and obtain their full value. Again, in case of accident and the dropping of the underframe upon the roadway, the bottom angles are bent or broken, making a difficult repair operation.

6 In general the deep side sill has been discarded because of the difficulty of inspection beneath the car. The deep center sill is much in vogue at present because it looks strong, but on a car with deep center sills inspection must be made of the parts attached to the underframe from one side of the car at a time, and the introduction of axle light equipment becomes difficult on account of the interference with the deep sills. Again, to sustain its own weight without deflection on a 60 ft. span, too much weight of metal is required to make such a sill economical.

7 Of the second type, that is, with the whole weight to be carried on the center sills, a common form (Fig. 2) has center sills of two special 18-in. channels with ½-in. cover plates top and bottom, all sections extending full length of the car in one piece. The box girder so formed has a square inch section of 50, and the superstructure load is transferred to these sills by means of four cross bearers, two of which take the place of the body end sills in other design. There are no side sills as such, the angles here shown simply forming the attachment for the superstructure. The parts are usually assembled with the bottom of the sills upward and allowed to deflect. The girder is then reversed and the camber straightens out by the weight of the

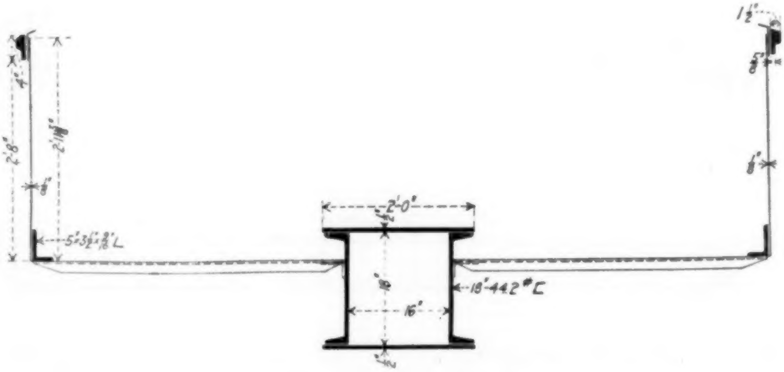



FIG. 2 TYPE *b*: WHOLE WEIGHT CARRIED ON CENTER SILLS

metal. The sills are the same depth and section throughout their entire length and with this construction a truck of special design must be used, the center plate of which must be nearer the rail than usual. The weight of the body rests upon the side bearings as well as the center plate. About 20 sq. in. of metal in the sides is available to help sustain the load. The service given by this underframe has been excellent.

8 The third type, with all the weight carried by the car sides has the center sills used only for buffing and pulling. An example shown in Fig. 3 has two I-beams running full length of the car in one piece, with a square inch sectional area of 23. They are held up by the three cross bearers which pass under and are attached to them. There are no side sills, the carrying members being the sides of the car. These members are composed of ⅛-in. plates, about 36 in. deep,

stiffened vertically by the window posts and having a 6 in. by 6 in. by $\frac{5}{8}$ in. angle at the bottom and an equal square inch section of metal at the belt rail, the two girders having a square inch section of 48 in all. With this construction a substantial body bolster is essential, as the load must be carried at the bolster extremities. Usually a cast-steel structure, built into the underframe and securely riveted to it, is used, the metal may thus be economically distributed. With an underframe of this type there is no trouble due to difficulty of inspection or interference with attachment for axle light or other equipment under the car.

9 The fourth type (Fig. 4) is a combination of types *b* and *c*. Here deep center sills are used, having a square inch section of, say

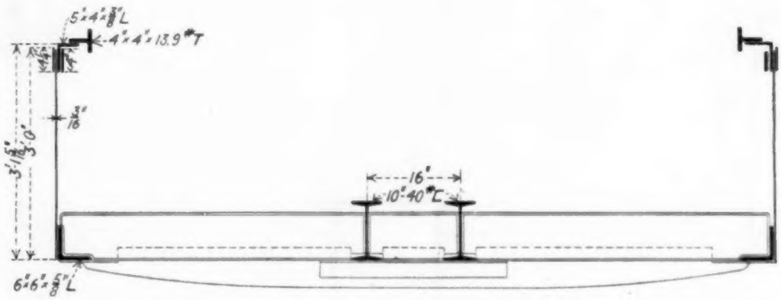


FIG. 3 TYPE *c*: WEIGHT CARRIED BY CAR SIDES, CENTER SILLS USED ONLY FOR BUFFING AND PULLING

40 at the center and 39 in cast steel at the draw gear. The side girders have a square inch section of 21 in the two. Most underframes of this type now in service are built with cast-steel end portions which include in one casting the body bolster, platform, side and center sills extending as far back of the bolster as may be necessary to secure a substantial connection to the center sills proper. This center member we do not consider as properly constructed for the reason that the section is unbalanced, an excess of metal being used on the top. Heavier angles or a cover plate should be used on the bottom, which would add about 10 sq. in. or more of metal.

10 The four types illustrated are of underframes actually in service. A comparison of cross-sections discloses the fact that no matter from what angle the designer has approached the problem, approximately the same square inch cross-section has resulted. If, therefore, any one type has an advantage in weight over the others,

it must be attributed to difference in the cross members of the underframe.

11 These four prevalent types have been recognized by the United States Government. The specifications of the Postoffice Department for the construction of steel postal cars provide as follows:

*a* Heavy center sill construction, the center sills acting as the main carrying member.

*b* Side-carrying construction, the sides of the car acting as the main carrying members, having their support at the bolsters.

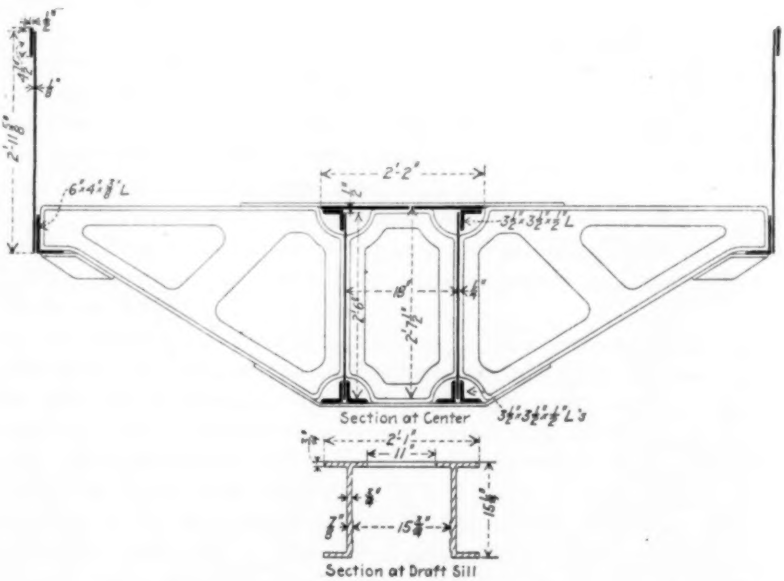



FIG. 4 TYPE *d*: WEIGHT CARRIED BOTH BY CENTER SILLS AND CAR SIDES

*c* Underframe construction, in which the load is carried by all the longitudinal members of the lower frame. The superstructure shall be of steel.

*d* Combination construction in which the side frames carry a part of the load, transferring it to the center sills at points remote from the center plate for the purpose of utilizing uniform center sill area.

12 While several of these types have been in service for a number of years the required time has not passed in which to develop structural defects due to unseen causes, such as fatigue of metal, crystallization, etc. If such defects exist they should make themselves known during the next three or four years, if freight construction is any criterion.

## No. 1388 *d*

# ROOF STRUCTURE FOR STEEL CARS

By C. A. Seley,[1] Chicago, Ill.

Non-Member

Roofs for steel passenger equipment cars are of two classes, the clear-story type with minor variations and the oval type. As regards contour and general appearance, they are the same as the long established standards for wooden cars, but varied as to constructive detail, due to materials employed.

2 The advent of the steel car has rather encouraged the use of the oval or round roof, as it is often called, particularly for cars used for baggage, express, and postal purposes. It is cheaper to build and maintain and fulfills requirements for such cars. For passenger cars the clear-story type prevails very generally, as it assists in lighting and ventilation and in decorative effect.

3 The framing for oval roofs consists of car lines, each a single member, bent to the shape of the arch and extending from plate to plate. There are no through longitudinal members and the roof sheets are riveted to the car lines.

4 Framing of clear-story roofs is of two general classes, one employing car lines of one piece extending from plate to plate and carrying the longitudinal upper deck sills and plates, and the other class an extension of the side framing posts as far as the upper deck sill. To these extensions are attached a member which comprises deck posts and upper deck car lines. It is difficult to approximate the strength of the more direct lines of the oval roof in the design of the clear-story roof, and all riveted connections must be thoroughly considered. The deck sills and plates are through members, act as end stiffeners, and add to the longitudinal strength.

5 The shape of the car lines of either type of roof should be such as to facilitate fastening of roof and of the inner ceiling or finish, and between these there should be a generous amount of insulating material to intercept the heat of summer and the cold of winter.

6 The committee of engineers who framed the specification for

[1]Mechanical Engineer, Rock Island Lines.

---

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

full postal car construction, which was approved by the Postoffice Department in March 1912, contains the following paragraphs in regard to the roofs of such cars and is probably as authoritative a statement as there is available. The strength of roofs of some cars that have been rolled over in accidents has been checked against the formula used, and it has been found ample to afford support against serious roof distortion in such cases.

7 The postal specification reads as follows:

ROOF

"General

The roof may be of either the clear-story or turtle-back type, depending on the standard contour of the railroad for whose service the cars are built. In the clear-story type, the deck plates shall be in the form of a continuous plate girder, extending from upper-deck eaves to deck sill, and either built up of pressed or rolled shapes or pressed in one piece from steel plates. The car lines may be either rolled or pressed steel shapes, extending in one length across car from side plate to side plate, or may extend only across upper deck. In the latter case the lower deck carlines may be formed by cantilever extensions of the side posts or by independent members of pressed or rolled shapes. In the turtle-back type, the car lines may be of either pressed or rolled shapes, extending in one length across car between side plate and side plate, or may consist of cantilever extensions of the posts.

"Car Lines

The projected area of the portion of roof in square feet, supported by car lines, divided by the sum of the section moduli of the carlines, must not be more than 100.

"Roof Sheets

Roof sheets, if of steel or iron, shall be of a minimum thickness of 0.05 inches, and either riveted or welded at their edges."

8 The design of the roof is also subject to the general paragraphs on stresses and details of the postal car specification.

9 There are several bills in Congress having in view the substitution of steel passenger equipment on railroads for present wooden cars. Should any of these become law, specifications for construction will be necessary, and, as the postal car specification has been approved and adopted as standard by the Government, no doubt this specification will be used as a basis in determining the requirements for other steel passenger equipment cars, not only for the roofs, but for the other features of construction.

No. 1388 *e*

# SUSPENSION OF STEEL CARS

By E. W. Summers,[1] Pittsburgh, Pa.

Non-Member

If we could operate steel cars over rails having no kinks, curves or irregularities in their alignment, in other words, over an absolutely straight track, there would be little need of springs or other devices for flexible support.

2 Unfortunately the roadways we have to contend with cannot be made or maintained in true alignment. Frost and water make constant changes in the track support. Lateral curvature requires super-elevation of the outer rail. In passing from a tangent to a curve, or vice versa, the tracks under one truck are in wind with those under the other one, sometimes as much as 4 or 5 in. depending upon the degree of curvature and the length of the car.

3 Steel car bodies of the enclosed type, such as box cars, mail, baggage, or passenger coaches, are of rigid construction and have high torsional resistance. A three-legged stool on an irregular floor surface will stand upon all of its legs while one having four legs may carry all of its load upon two diagonal supports.

4 The use of truck springs helps the illusion that we are distributing the car body load on all of the wheels. The uneven deflection of the springs indicates directly the increased load of one spring over the other. When the track surface is warped more than the total spring travel, the whole load is carried at two diagonal corners, tending to twist the car body. This twisting tendency is constantly changing, first in one direction and then in the other, as the super-elevated rail changes from one side of the track to the other. The effect upon wooden passenger cars is to work the joints loose and cause them to screech and grind like the spokes of a wooden wagon wheel in hot dry weather.

5 The side bearings of steel sleeping cars pop like sledge hammer blows when the car is taking or leaving a curve. The slight twist in the track surface throwing excessive load upon two diagonal corners

[1]President, Summers Steel Car Company.

---

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

of the car causes the bearings to grip and adhere to each other coincident with the slewing of the truck. When the twisting of the truck exceeds the play in the parts around the truck bolster the side bearings let loose and jump with resulting hammer blows. More efficient roller side bearings may prevent the gripping and jumping, but the uneven load is still present. The twisting effect upon the car body is not removed.

6 Failure in roofs of wooden box cars and the resulting damage to merchandise in transit is due to this constant twist. Roof designers have attempted to remedy this by making the roof flexible and with slip joints. To be consistent they should go further and make the whole car of india rubber. A practical construction for the enclosed type of steel car bodies must and always will be rigid and of high torsional resistance.

7 The necessity for flexibility between the car body and the trucks, and for an even distribution of the load upon all of the wheels seems not to be fully appreciated as yet, but with each succeeding year wrecks due to broken rails, wheels and truck structure will drive this home. Suspension of steel cars, as has been developed by the writer in the past three years, does permit of a more even distribution of the load upon the wheels than with center-bearing trucks.

8 Fig. 1 is an illustration of a cross-section through an engine tender at the center of one of the trucks. It illustrates the method of suspension referred to and is applicable to any kind of car.

9 The inclined hangers *a*, the cradle *b*, and the side rockers *c* are shown heavily shaded. There are two inclined hangers at each side of each truck. A heavy rectangular bar extends through the lower ends of the hangers *a*. A cast-steel bracket, which is part of the car underframe, rests upon each end of the rectangular bar. The upper ends of inclined hangers *a* are supported upon the outer end of the cradle which rests upon the segmental rocker *c* and transmits the car body load directly into the truck side frame. The lower ends of hangers *a* are maintained a fixed distance apart transversely of the car, by reason of the brackets *d* being a fixed part of the car underframe. Their upper ends are held at a fixed transverse distance by their connection with the cradle *b*. Both the upper and lower ends of bars *a* are pivotally connected with rolling contact.

10 With one end of a car on level track and the other end having one rail at a higher elevation, the tendency will be for the high rail to carry all of the load at that end of the car, or to have the car support taken at two of its diagonal corners.

11 With *inclined hanger suspension* the car will swing sideways, the hangers at the high rail swinging inward and downward, while the ones at the other end of the cradle swing outward and upward, picking up the load at the low rail and maintaining its distribution on all of the wheels much the same as if suspended by two bars from

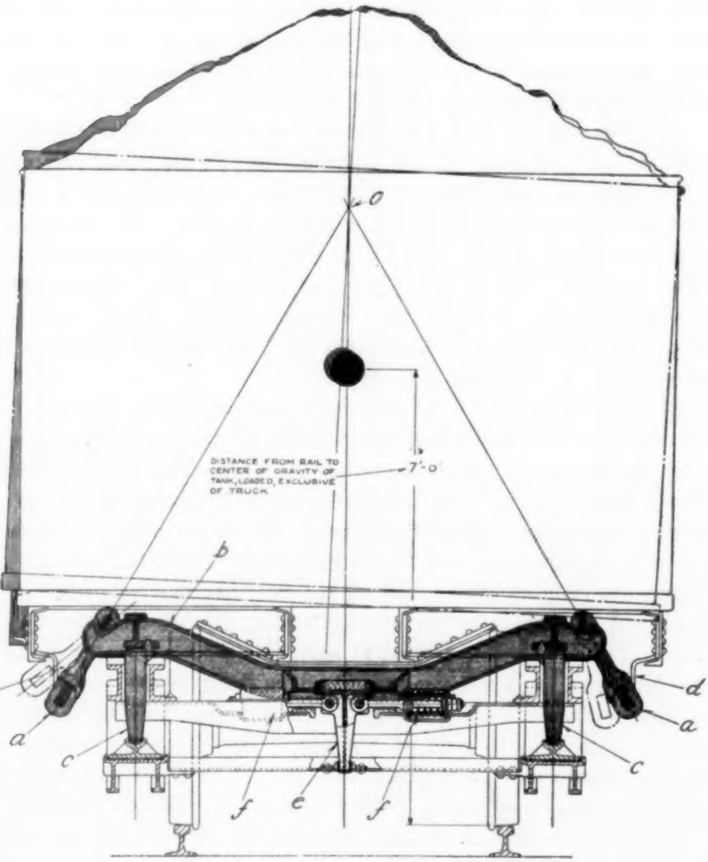



FIG. 1 CROSS-SECTION OF ENGINE TENDER AT CENTER OF ONE OF TRUCKS

a point $O$ at the intersection of the center line of the inclined hangers extended. The load or rigid car body will find its own position vertically under this common point of support $O$, each of the extended suspension bars taking its share of the load.

12 The slight warping of the track surface, which causes all of the load to be carried on half of the wheels at two diagonal corners of a rigid car body with the ordinary center-bearing truck, is corrected by the short inclined suspension bars *a*, practically the same as if the suspension was from point *O*.

13 It is the inclination of these bars that makes vertical adjustment possible, one bar swinging inward and downward, the other swinging outward and upward at one end of the car, the bars at the other end swinging in the opposite direction, the car body finding its position much the same as a boat does in water.

14 Imagine the bars *a* at one end of a car swung to the left, as shown in the unshaded dotted position, and at the other end swung to the right an equal amount: this makes correction for a warped track surface of about 8 in. in the length of a car. Or, imagine the hangers *a* swung to the left at both ends of the car, as shown in the dotted position: this is the inclination the car body will assume in rounding a sharp curve at high velocity, the top of the car leaning inward and the bottom swinging outward, the position assumed by a bicycle rider in rounding a curve.

15 The cradle *b* is pivoted about a vertical axis on the king pin *e* and can also have movement transversely of the car, this movement being limited by the action of springs *f*. On account of the inertia of the car body and its load, the cradle moves transversely of the car, rotating the hangers about their lower ends when rough track is encountered at high speed. Without this cradle movement, the inclined hangers are impracticable; with it, the car body movement is without jar or jerk and we have perfect adjustment for all track conditions.

16 The car body is carried at each side almost directly under its rigid side girders, which by position have great depth and can carry the load with the least deflection. Floor beams may be made continuous from side to side of the car. The necessary buffing and tugging column may be disposed with its web in a horizontal position under the transverse beams, greatly simplifying the car framing.

17 With the advent of steel construction for enclosed cars a rigid structure came into use, one that cannot be handled over rough track as we have been handling the spongy wooden structure. There may be much hewing and chopping into old methods before the necessary compromise is made between the rigid car body and the changeable track surface, but why not do it all at once, and stop fooling with dynamite?

No. 1388 *f*

# SIX-WHEEL TRUCKS FOR PASSENGER CARS

By John A. Pilcher, Roanoke, Va.

Member of the Society

Consideration of the subject of trucks for steel passenger cars is practically a consideration of trucks for any passenger car, the primary thought being that steel passenger cars should have steel trucks to prevent the possibility of fire, and also because of their great weight, metal is the most suitable material for strength and durability that can be used in the limited space available for the truck. The fire damage from a wooden frame truck could not be serious on a steel car, and there are wooden cars equally as heavy as the general run of steel cars; the writer having one in mind in the construction of which the sills were plated with 8-in. channels, weighing 172,700 lb. A few steel cars weigh as much as this, but we have no record of any weighing more. However, for steel passenger cars we will consider only the all-steel truck.

2 Practice of the past brings to our attention the pedestal type of passenger truck construction both for four-wheel and six-wheel trucks, the general characteristics of both being identical. The six-wheel truck with the same size axle is, of course, capable of greater load and also of transmitting to the car the track irregularities to a less extent, because the results of the irregularities are modified by the system of equalization. In the six-wheel truck the location of the equalizer springs is fixed at a definite point between the wheels.

3 While the details of these two trucks differ slightly, their functions are practically identical. Both trucks have been used for a considerable length of time, but the four-wheel truck was evidently developed first and its necessary functions, determined by experience, were later incorporated in the design of the six-wheel truck, which was probably first brought about by the increased loads.

4 Except for the especially constructed truck used by the Pennsylvania Railroad and one other, which we understand has been designed, these are the only regular types of trucks available.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

5 *Wheels.* For passenger service, the wheels have been practically narrowed down to steel tired wheels and wrought-steel wheels. The steel tired wheels have been of many forms of centers and fastenings; the latest recommended practice of the Master Car Builders' Association is that the tire be shrunk on and bolted. The recent development of the solid wrought-steel wheel has made available for passenger car service a wheel equally as safe and durable as the steel tired wheel at a very much reduced cost. The Master Car Builders' recommendations recognize both the 36-in. and 38-in. size in this wheel for passenger service, the 36-in., however, being the most generally used. These wheels, if carefully turned, should give as satisfactory service as any wheels available.

6 *Axles.* The standards of the Master Car Builders' Association gives the choice of selection of four sizes of axles:

| Size of Journal | Axle Load, Lb. |
|---|---|
| 3¾ in. x 7 in. | 15,000 |
| 4¼ " x 8 " | 22,000 |
| 5 " x 9 " | 31,000 |
| 5½ " x 10 " | 38,000 |

7 They also offer an axle as recommended practice with 6 in. by 11 in. journals for 50,000-lb. axle load. These loads, however, are for freight service; for passenger service we would recommend the use of from 60 per cent to 75 per cent of the loads used in freight service, based on the light weight of the car, and limiting the load to about 90 per cent of that in freight service, considering the weight of both car and lading. The lighter rating is, of course, to be taken for cars such as baggage and express, since the increased weight on account of lading would be heavier, while the higher rating could be taken for coaches and similar cars where the increase of the lading would be light. Table 1 gives the sizes of axles, and relative light weights of cars on this basis.

8 The Postoffice Department has limited the maximum load per wheel for postal cars to 15,000 lb. when using 5½ in. by 9 in. journals, and to 18,000 lb. when using 5 in. by 10 in. journals, making a further limitation based upon 18,000 lb. as the maximum brake load for any one cast-iron brake shoe under emergency conditions of brake application. This limitation of wheel loads, after deducting the weights of the wheels and axles, allows a pressure of 304 lb. per sq. in. projected area on the 5 in. by 9 in. journals, and 300 lb. per sq. in. projected area on the 5½ in. by 10 in. journals, also a pressure of 1522

lb. per lineal in. on the 5 in. by 9 in. journals, and 1665 lb. per lineal in. on the 5½ in. by 10 in. journals, and from the experience that some roads have had these seem to be just as high as should be allowed.

TABLE 1 SIZE OF AXLES AND WEIGHT OF LIGHT CARS

| Axles, In. | Four-Wheel Trucks, Lb. | Six-Wheel Trucks, Lb. |
|---|---|---|
| 3¾ x 7 | 40,000 to 52,000 | 60,000 to 78,000 |
| 4¼ x 8 | 52,000 to 72,000 | 78,000 to 108,000 |
| 5 x 9 | 72,000 to 100,000 | 108,000 to 150,000 |
| 5½ x 10 | 100,000 to 120,000 | 150,000 to 180,000 |

9 *Boxes and Contained Parts.* The Master Car Builders' Association has provided standard passenger boxes for axles with 3 ¾ in. by 7 in., 4 ¼ in. by 8 in., and 5 in by 9 in. journals. For the 5 ½ in. by 10 in. journal, which is often in use, they have not yet established recommended practices, but the previous designs are having their influence on the shape of the box for this journal.

10 *Pedestals.* Cast-iron pedestals seem to be usually the accepted material, and the Master Car Builders' Association has also provided standards to suit the boxes.

11 *Equalizer Springs.* These are four in number on both the four-wheel and six-wheel trucks, and while necessarily provided with a limited amount of deflection, they relieve the heavy truck frames of shock, and on six-wheel trucks provide the points of support for the proper equalization.

12 *Wheel Pieces or Side Frames with Transoms or Cross Ties.* These constitute the truck frame to hold the other parts in their relative position, and at the same time transfer the load from the bolster hangers to the equalizer springs. Being structures supported at four points, they necessarily have to be supported on springs to prevent excessive stresses due to any variations in the height of these four points. As an illustration, when the truck on a tangent is approaching a curve the rise of the outer rail is about 1 in. in 50 ft. This will raise one of these four points above the plane passed through the other three, and, while the difference is small in the short length of the truck, the irregularity has to be taken up by the springs, otherwise the truck frame would be similar to a four-legged table with one high leg.

13 When we consider the case of a derailment where one wheel of the truck, whether four-wheel or six-wheel, falls into a deep hole, or drops from a high rail, we find this condition exaggerated to such an extent that the whole load will be supported on two points. Then unless the structure is sufficiently flexible to follow, it will necessarily have to be strong enough to resist this abnormal load.

14 The calculation of the stresses under such uncertain conditions of loading is certainly a very complex problem. It is a pertinent question whether or not the designers should undertake to care for such an abnormal condition.

15 *Bolster Hangers.* The lateral movement of the bolster, one of the very necessary features of a passenger truck, is usually accomplished by the use of swinging hangers. This movement should be limited to from $1\frac{1}{2}$ in. to $1\frac{3}{4}$ in. each side of the center, and in placing this limit arrangement should be made so that the stop will not be abrupt. This is ordinarily accomplished by the use of short hangers, or when long hangers are used by the addition of lateral motion springs, either of which offers an increasing resistance. Rollers on cylindrical or curved plains can produce the identical movement made by the short hangers.

16 *Bolster.* On the four-wheel truck the bolster is a simple beam, but on the six-wheel truck we have a more complex structure resting on four points of support. This condition brings up the same complex problem referred to in connection with the truck frame supported on four points, except that it rests on much more flexible springs than does the truck frame. These springs can hardly be expected to take up all of the variations in elevation that will likely be met with in case of a partial derailment. The same question as to whether or not the designer should allow for such abnormal conditions is again raised.

17 *Center Plate.* The usually accepted center plate for passenger cars is of the spherical pattern, allowing more perfect adjustment, and more even distribution of weights than can be obtained from the flat bottom center plate, but making necessary close and accurate adjustment of the side bearings to prevent the rocking movement between the car body bolster and the truck bolster.

18 The frictionless center plate would of course be very desirable, but conical rollers and balls of sufficient number, of the size that can be put in the available space, seem not to have been as successful as

could be wished. The ingenious designer is still at work on this particular problem.

19 *Side Bearings.* Side bearings must be made so that they can be readily kept adjusted to reduce to a minimum the rocking movement between the car body bolster and the truck bolster, and in this way confine the oscillation of the car to the variation in the deflection of the springs on either side.

20 The relative location of the side bearings, each side of the center, is a question often discussed. In passenger cars the practice generally is to place them at as great a distance from the center as practical. This in our judgment is correct, and of particular advantage in the case of frictionless or roller side bearings.

21 Where the side bearings are in actual contact and the bolsters are rigid, the oscillation of the car is controlled entirely by the difference in deflection of the springs on either side, so that if the side bearing is set out sufficiently far to prevent the car body upsetting on the truck, it serves its purpose in preventing car oscillation as well there as at any other location.

22 For the same type of side bearing, it offers just as much, but no more, resistance to turning than if located far from the center, because as the lever arm is increased the pressure is reduced in like proportion.

23 When the car on a tangent is approaching a curve, the rise of the track on the outer rail tends to bring a pressure on the side bearing of the leading truck, next the outside of the curve, and on the side bearing of the trailing truck toward the inside of the curve. Where the side bearings are in contact this variation in elevation has to be taken care of by the deflection of the springs which have to deflect the same amount whether the load is exerted on the bolster, at a point near the center, or far away from the center. If the load comes far from the center it takes much less pressure to influence the deflection of the springs. This would be to the decided advantage of the side bearings, particularly in the case of the frictionless side bearing, in preventing wear and would also, to a more limited extent, be of advantage to the ordinary flat side bearing.

24 *Brakes.* On passenger cars, the pressure on the brake shoes approximates the loads on the wheels. Particularly is this the case of coaches where the lading is only a small proportion of the total weight. In some braking arrangements the brake shoe load is even greater under certain conditions than the wheel load; therefore the lighter the wheel loads the better for the brakes. This is a decided

argument in favor of the six-wheel trucks for heavy cars, and an argument against the use of four-wheel trucks under heavy passenger cars, even though the weights can be readily sustained by the use of sufficiently large axles.

25 The application of the brakes to the six-wheel trucks in such a manner as to allow for the adjustment of worn shoes and worn wheels is a very difficult task on account of the limited space available. It is almost impossible to accomplish this task with the use of wheels less than 36 in. in diameter.

26 *Six-Wheel Trucks.* Since steel cars are of recent construction, and recent conditions have generally called for large cars, the weight is almost always great. The six-wheel, all-metal truck has the following advantages which make for its selection over other types:

*a* It is non-inflammable.

*b* It provides a strong material to resist the heavy loads, and occupies only a limited space.

*c* It provides a durable material.

*d* It reduces the axle loads, and the unit load on the bearings, lessening the liability to hot boxes, reducing the pressure on the brake shoes, lessening the tendency to heat the wheels and shoes, adding to the life of the brake shoes, and reducing the frequency between renewals and adjustments.

*e* It spreads the heavier loads over a greater area of structures, and brings more points of contact with the rail, reducing the influence of track irregularities on the riding of the car, and in cases of very heavy cars, where the unit pressure between wheel and rail might approximate the elastic limit, reduces the tendency to shell the wheel and roll out the rail, adding to the life of both.

27 It has been estimated that for a passenger car making 50,000 miles per year, the cost for hauling the car is 5 cents per lb. per year. If the six-wheel trucks weigh 14,000 lb. per car more than the four-wheel trucks necessary to carry the same car, it means the hauling of 14,000 lb. additional at a cost of $700 per year, which brings up a question for vital consideration.

28 While the wheels, brasses, and brake shoes, and other such removable parts may individually have a longer life, there are also more of them in service during the period. Careful comparison would have to be made to determine which has the advantage at this point.

29 *Four-wheel Trucks.* The four-wheel, all-metal truck is also available in connection with steel cars, and has the advantage of reduced first cost, reduced weight, smaller number of parts to maintain, and if the car is sufficiently light for the unit stress between the rail and wheel to be kept down to a point well below the elastic limit of the material, they should be given serious consideration. The only drawback under these conditions is the possibility of its reduced riding qualities. Its decided advantage in reducing the weight of the train should help to make it a favorite because of the corresponding reduction in the cost of transportation.

30 *Cast-Steel vs. Riveted Wrought-Steel Frames.* The introduction of heavy passenger equipment is rapidly doing away with both the four-wheel and six-wheel wooden frame trucks. The reduced cost of maintenance amply justifies this change if our information is correct. Cast-steel one-piece frames, and riveted wrought-steel frames of various cross-sections have been worked out and are now in use; both are reported as giving satisfactory service, but figures showing the exact relative cost of maintenance are not available.

31 The cast-steel one-piece frame has become a great favorite even in the face of the high unit cost of these particular castings. The adaptability of the castings to the various changes of form and section necessary on account of the limited available space has no doubt had much influence. The attractiveness of the one-piece structure, eliminating all joints, and furnishing a frame ready set up, is another strong argument in its favor. The manufacturers having control of this cast-steel truck frame have evidently been successful in reducing to a minimum the concealed flaws often met with in steel castings. This, no doubt, has added largely to its popularity.

32 While the absence of riveted joints and the consequent doubling of material at the joints, helps to keep down the weight, the fact that the working fiber stress of cast steel is taken low, and the sections at many points have to be made larger than is necessary on account of foundry limitation, the weight of the frame as a whole is great. This added to the large unit cost for special steel castings makes the user pay well for the advantages gained.

33 The riveted wrought-steel frame seems to have been held back in its development by the success of its rival in cast steel. Many users have shown conservatism in making use of the good thing already considered acceptable, hesitating to try out the different construction with the hope of lower first cost, with less weight, and equally good service.

34 Wrought steel at a very moderate unit cost has the advantage of being a very reliable material which can be worked to a relatively high fiber stress. The cost of fabrication, when the work is done in any large quantity, when added to the cost of material, will still leave a large margin in its favor. Is it possible that the lack of an especially interested advocate has prevented its virtues from becoming prominent, and delayed the experience needed to prove, in actual service, its worth?

35 We find that practically all of the prominent car builders have already worked up designs for wrought-steel trucks, and are ready to construct them if the purchaser so desires, but they do not seem inclined to push them, as they evidently offer no special inducement to their own advantage. Only a few have been built and placed under cars by them, and in some cases none, but from what I have been able to find out they have confidence in them.

36 I find several railroad companies building and using both four and six-wheel trucks, of the usual type of construction, with riveted wrought-steel frames, and from all reports they are giving satisfaction.

37 Another prominent railroad is using both four and six-wheel trucks, of a form of construction differing from the ordinary type, built of riveted wrought steel. As a large number of these are in daily evidence, and are constantly being built by them, they must be proving the worth of the riveted wrought-steel construction, as well as that of the special type of construction.

38 Experience of several years and careful comparison of the cost of maintenance will be needed to say whether the one-piece cast-steel frame, or the riveted wrought-steel frame truck will be the most advantageous, when both the first cost and weight are considered along with the cost of maintenance.

39 Variety of choice offers an opportunity for discussion. In the hope of bringing out this discussion we advocate for steel passenger cars: (*a*) Six-wheel truck; (*b*) the riveted wrought steel frame; (*c*) the use of the Master Car Builders standard axles, boxes and parts, and pedestals; (*d*) 36-in. wrought-steel wheels.

No. 1388 *g*

# STEEL INTERIOR FINISH FOR STEEL PASSENGER CARS

By Felix Koch,[1] McKees Rocks, Pa.

Non-Member

Every one who has followed the progress in steel passenger car construction during the last ten years, which is about the age of the oldest steel passenger car, has noticed that very little, if any steel was used in the interior finish until within the last four or five years.

2 The first attempt to use steel in passenger cars resulted in steel underframes with wood superstructure. The next development provided steel underframe and steel superstructure, but with wooden roof and wood interior finish. Further developments eliminated the wooden roof and the final efforts produced an all steel car. Considering that this development was made during a period of four years, the results obtained are, to say the least, highly gratifying.

3 The earlier designs of steel cars with steel interior finish are sometimes called all steel cars, leaving the impression that they are fireproof in every respect, but this is not correct because too much wood was used in the form of wood furrings to enable the application of the steel finish with wood screws. These furrings were, of course, not exposed to view, but they nevertheless placed the cars outside of the classification "all steel cars." The idea that it was necessary to use wood furrings in order to make it possible to apply steel finish, or in other words, that wood screws had to be used, machine screws not being considered practicable, accounts to some extent for the tardiness in the introduction of steel in the interior finish.

4 The earlier specifications and designs for steel passenger cars made the use of machine screws for applying the interior finish prohibitive and impossible, which, of course, made it necessary to employ other means such as bolts or wood screws. Bolts for this purpose must have heads of special design to allow their insertion through slotted holes, etc., and to prevent them from falling through during the application of the nuts. The nuts, being exposed, are objectionable as they give

[1]Assistant Mechanical Engineer, Pressed Steel Car Company.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

an unsightly appearance, even if special cap nuts are used in place of the ordinary nuts, besides there are many places on a car at which it is impracticable to apply bolts. Therefore, to avoid machine screws and bolts the space between the outside sheets and the interior finish was filled with wood furring to allow for the use of wood screws. The objections to machine screws, caused by the belief that they would work loose in a short time, has, however, disappeared from experience gained through actual service as it has been shown that if set in white lead and properly applied they are entirely reliable.

5 There has always been and there still is a great difference of opinion as to how far it is advisable to substitute metal for wood in passenger car construction. The use of a small amount of wood in the interior finish, as for instance window sash moldings, seat arm rests, window capping, etc., should not be objectionable as it has certain advantages over steel which are desirable, but wood is used for such details to a considerable extent, and hundreds of cars are now in service in which the small amount of wood used in the interior finish cannot be detected except by an expert and such cars are to all intents and purposes fireproof cars, but the aim of many designers has been to eliminate the wood wherever possible on account of the many advantages possessed by steel, among which may be mentioned:

*a* Steel finish means non-combustion in case of fire.

*b* Steel prevents splintering in case of wreck.

*c* Steel finish can be easily removed should it become necessary to repaint the car at the inside surface of steel sheets, as the life of the steel car, to a certain extent, depends on the condition of the paint.

*d* Steel finish makes it possible to increase the interior width of the car where outside width is limited. This has been found particularly valuable in designing subway, elevated or suburban steel passenger equipment cars.

*e* Steel finish will avoid trouble which may be experienced due to different expansion of materials, steel against wood. This point need not be considered with steel and makes it unnecessary to provide for relief in all members of the finish running longitudinally, such as upper and lower deck sill moldings, etc. In fact, the steel finish has revolutionized to some degree the designs of wood finish in the wooden cars built since steel cars came in vogue. The cars of today are built on more sanitary lines, and

fancy moldings, fretwork and carvings have disappeared without losing sight of giving the cars an artistic finish, avoiding thereby lodging and breeding places for all kinds of germs which the world is fighting against today.

*f* Steel finish will, by comparison, be cheaper every year for the reason that it becomes more difficult to obtain the right kind of lumber for interior finish, which, of course, means increase in price of wooden cars.

*g* It is continuously becoming more difficult to obtain men who have had sufficient experience in applying wood interior finish, whereas it does not take the same experienced men for applying steel finish. A man requires from three to four years' apprenticeship to become an expert able to apply wood finish to a car, whereas an average intelligent man who is familiar with tools is able to become an expert in finishing cars with steel finish in from six to twelve months, and this fact of labor will have to be taken into account sooner or later.

*h* A more uniform color can be maintained on steel finish than on wood which comes in different shades, and it is very difficult and expensive to match perfectly all parts in one car with regard to shade without additional expense of glazing. Furthermore, the average life of paint applied to steel finish will be much greater than to wood finish for the reason that wood darkens with age. This, of course, influences the paint which is a disadvantage from the standpoint of illumination. Should it become necessary to repaint a car of wood finish, reworking of the finish by removal of the varnish and scraping is necessary, whereas in the steel finish the scraping is eliminated and the removing of varnish is alone required to be able to repaint the car.

*i* Steel finish is of advantage from a building standpoint in the handling and working up of material to make ready for application. Steel details can be worked up to a large extent before they are applied to the cars, which make it possible to manufacture the interior finish in much less time by the use of more men than it is possible to employ when applying a wood finish, as only a limited number of men have room to work at the same time in a car when the greater part of the fitting and cutting, etc., has to be done.

This has facilitated the establishment of a number of manufacturing concerns who devote their efforts almost exclusively to producing steel interior finishes not only for passenger cars but also for buildings. In addition to these any manufacturing company equipped with the necessary machinery for the making of drawn moldings, breaker presses, and ordinary welding and spot welding machines, is able to handle this class of work for railroads or carbuilders, who may not have the necessary equipment to do the work in their own shops and prefer to buy the interior finish as they buy other specialties.

6 All of these advantages are almost exclusively confined to the use of steel or other metals, although a composite material of a wood pulp nature or similar material made fireproof and waterproof by different processes, if applied in a proper way and used for ceilings and below the window sills, is not objectionable, and it may be applied in practically the same manner as steel.

7 The advantages possessed by wood over metal as a non-conductor can be very much reduced by the use of proper insulating material correctly applied. The use of proper insulation is of course of great importance and manufacturers of that class of material as well as railroads and car builders are giving a great deal of attention to the subject, and the time does not seem to be far distant when steel cars with interior finish of wood will be as scarce as steel passenger cars were ten years ago.

8 A great deal more could be said on this subject, but it is hoped that what has been brought out will show that steel interior finish has certain advantages not possessed by other material commonly used in passenger cars and that the disadvantages are few and not insurmountable.

No. 1388 *h*

# PAINTING OF STEEL PASSENGER CARS

By C. D. Young, Altoona, Pa.

Member of the Society

A fundamental reason for painting any surface of a passenger car is to protect it from the damaging effects of the air which is more or less loaded with gases and moisture. For example, oxygen is destructive of iron and steel and when sulphurous gases are present they are quickly oxidized into sulphuric acid which is very corrosive to unprotected metallic surfaces. It, therefore, becomes necessary to protect the surface by a covering, and paint forms a substantial and convenient means for accomplishing this. If properly made and applied, it is an impervious coating, affording the needed protection by forming a hard waterproof, rubber-like sheeting or film which has sufficient elasticity to conform itself to the contraction and expansion of the surfaces to which it is applied. In addition to protection the surfaces may be beautified and embellished by the proper selection of pigments so as to bring about the harmonizing and artistic effects desired.

## WOODEN EQUIPMENT

2 The painting of wooden passenger-car equipment has been, in the main, successfully accomplished, the painting schedule for the outside is briefly as follows: Apply two coats of primer, putty and glaze, followed by three or four coats of surfacers, as found necessary, after which the surfaces are rubbed down smooth with emery and oil, when two coats of shade color are put on. The necessary striping and lettering follows, completing by three coats of finishing varnish, consuming in all about sixteen to eighteen days.

3 The finishing of the interior of wooden cars generally has been in the natural wood, consequently it is only necessary to prepare the surface for the varnishing. A representative schedule which is used is as follows: One coat of filler, in paste form, which is sandpapered down to a smooth finish. Add one coat of rubbing varnish and rub down with sandpaper, after which apply three coats of rubbing varnish,

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

and complete the finish, cutting down the gloss by rubbing with pumice and oil to produce the most pleasing "flat finish."

4 This method of finishing the wooden surfaces of cars has been attained with good results, so that naturally when the change to steel passenger equipment came some six years ago, a desire to retain as much past practice as possible seemed desirable. It was realized, however, that the all important point in the painting of iron or steel surfaces was to have the surfaces first thoroughly cleaned and entirely rid of scale and rust, as this is as necessary as the painting itself. To accomplish this, sand-blasting, where possible, was resorted to, supplemented by the use of wire brushes and emery cloth in the more obscure places and the more uneven surfaces. The sand-blasting, however, was confined largely to the outside surfaces and the latter practices to the inside portion of the car.

5 Iron and steel, while not presenting to the eye the same porous condition as wood, is full of finely divided pores, and the same atmospheric influences which enter the pores of wood and cause it to decay are ever ready to attack the unpainted surfaces of iron and steel, in fact the metal surfaces more readily combine with the oxygen and moisture of the air, forming what is rust or oxide of iron. Therefore, immediately after the sand-blasting and cleaning of the surfaces should come the application of the first or primary coat, as this is the most important one, from the preservative standpoint.

6 In the selection of a suitable primer it seemed but natural for the painter to be guided by the experience gained in the painting of locomotive tenders, and to follow the initial coats with practically the same process as with wooden cars, and I believe that so far as the subsequent coats are concerned, this practice was generally carried out by the earlier painting of steel passenger equipment. It is thought that an error has been made in this general practice, as will be explained later.

### STEEL EQUIPMENT

7 The schedule for painting steel passenger car trucks, underframes and superstructures is as follows:

8 *Trucks.* Before assembling, all surfaces on truck parts throughout, including all concealed surfaces, but not including wheels and axles, must be covered with one coat of suitable primer. After assembling, all surfaces (except wheels) exposed to view after the body of the car has been placed on trucks, must be covered with two coats of truck enamel.

9 *Underframes.* During the process of construction, all parts of the underframe, including concealed surfaces and surfaces where metal bears on metal, must be covered with two coats of good metal preservative of a non-inflammable nature. All accessible surfaces must be covered with a third coat of metal preservative.

10 *Superstructures.* Before assembling, all parts made of iron or steel, including the roof, must be covered with one coat of primer. A second coat of primer properly thinned with turpentine, or similar material, must be applied to all surfaces, including those which are concealed when the car is completed. Wherever possible, this second coat must be put on after the sheets are in place.

11 After assembling, the outside of side and end sheeting, including letter plate and deck plate, must be covered with one coat of surfacer, the rough and uneven places glazed with "surfacer composition," four coats of surfacer being added, rubbed down with linseed oil and emery cloth, two coats of desired color material added, followed by striping and lettering, then finished with three coats of finishing varnish. The outside of the roof must be finished with one coat of heavy protective paint, followed by one coat of a mixture composed by volume of three parts of mixed ground color and one part of the protective coating used. The top surface and edges of headlining should be painted with two coats of some preservative, or color paint.

12 The interior of cars should receive very careful attention in order to produce the desired finish. To illustrate fully the various steps and time taken to complete the painting, the following is given as outlining the progress of the work. This is attained with the use of surfacers, colors and varnishes containing a relatively large amount of artificial driers and varnish gums, in order to obtain the artistic finish desired for the interior.

HEADLINING

| | |
|---|---|
| 1st day | Apply one coat and stipple after application. |
| 2d day | Stand for drying. |
| 3d day | Apply one coat and stipple after application. |
| 4th day | Stand for drying. |
| 5th day | Apply one coat and stipple after application. |

SIDES ABOVE WINDOW SILLS AND ENDS

| | |
|---|---|
| 1st day | Apply one coat or priming. |
| 2d day | Stand for drying. |
| 3d day | Apply one coat surfacer. |
| 4th day | Necessary puttying and glazing. |

| | |
|---|---|
| 5th day | Apply as many coats surfacer as are necessary to make a level surface. |
| 6th day | Same as 5th day. |
| 7th day | Rub down with emery and linseed oil. |
| 8th day | Apply one coat of ground color. |
| 9th day | Apply one coat of ground color. |
| 10th day | Apply one coat of ground color. |
| 11th day | Apply one coat and stipple after application. |
| 12th day | Apply one coat rubbing varnish. |
| 13th day | Stand for drying. |
| 14th day | Apply one coat rubbing varnish. |
| 15th day | Stand for drying. |
| 16th day | Apply one coat rubbing varnish. |
| 17th day | Stand for drying. |
| 18th day | Rub with oil and pulverized pumice stone. |

SIDES BELOW WINDOWS

| | |
|---|---|
| 1st day | Apply one coat or priming. |
| 2d day | Stand for drying. |
| 3d day | Apply one coat surfacer. |
| 4th day | Necessary puttying and glazing. |
| 5th day | Same as 6th day. |
| 6th day | Apply as many coats surfacer as are necessary to make a level surface. |
| 7th day | Rub down with emery and linseed oil. |
| 8th day | Stand, awaiting bringing up other work. |
| 9th day | Stand, awaiting bringing up other work. |
| 10th day | Apply one coat bronze green. |
| 11th day | Apply one coat bronze green. |
| 12th day | Apply one coat of rubbing varnish. |
| 13th day | Stand for drying. |
| 14th day | Apply one coat of rubbing varnish. |
| 15th day | Stand for drying. |
| 16th day | Apply one coat of rubbing varnish. |
| 17th day | Stand for drying. |
| 18th day | Rub with oil and pulverized pumice stone. |

13 Formulae and panels for the various shade should be furnished the painters for their guidance in obtaining the shade of any of the colors which are desired.

RESULTS OF AIR DRYING PAINTS ON STEEL

14 The artificial driers and gums used in hastening the time of drying and hardening of the various coats and permitting the necessary rubbing continue this action so that the paints and varnish increase in hardness and brittleness, rendering them susceptible to

cracking and chipping, and the process of disintegration is aggravated by excessive expansion and contraction of the steel surfaces as compared with wood. The linear expansion of steel being more than twice that of wood would seem to indicate the use of more elastic coatings than formerly used for wooden cars.

15 This fact has been borne out in the service of the paint in a great many cases in an investigation which recently came under my observation. It was noticed that when some of the equipment had been in service about four months, the interiors of the cars were showing varnish cracks and checks. As time went on more cars gave evidence of this deterioration, the final outcome being that an investigation was made to see how serious the condition was. Some 400 cars were carefully examined, special attention being given to the selection of cars built by various manufacturers, where different makes of surfacers and varnishes were employed. An endeavor was also made to determine whether the cracking of the painted surfaces was confined to the varnish coats or the surfacer coats, or both.

16 In order to classify the various conditions found, four readings of percentages were arbitrarily taken, the condition of a new car being taken at 100 per cent:

| Per cent | Condition of Varnish and Surface |
|---|---|
| 90 to 80 | Good, no checking |
| 80 to 70 | Fair, slight checking |
| 70 to 60 | Medium, considerable checking |
| 60 to 50 | Poor, checked from outside varnish coat to metal |

Sample cars were selected to illustrate these various classes, and photographs were taken of the different defective surfaces so as clearly to indicate to the eye what the different percentages meant.

17 The result of this examination showed that the exteriors, including the sides, ends and vestibules, were in fair condition. There were a few exceptions to this, but they amounted to less than 6 per cent of the total having serious varnish and surface cracks. Interiors were found generally to be in a poor condition. About 80 per cent of the equipment examined had the varnish checked through to the surfacer.

18 Some of these conditions developed after four to eight months' service, indicating either that an entirely new system of painting would be necessary to overcome these troubles, or that a more elastic paint would have to be used for interior finishing under the present existing practice of painting steel.

19 To obtain some data indicating what should be done to meet the conditions, preliminary experiments were made by painting a number of panels and baking them in a heated oven. Repeated experiments along this line indicated that artificial driers could almost, if not entirely, be eliminated in the paint formulae and that more elastic materials could be used without the aid of artificial oxidizing agents. It was also observed that the elastic varnish used on the exterior of the cars could, under this system, be used to advantage on the interior, and by the aid of the heat of the oven they could be dried to the desired hardness, permitting the rubbing with oil and pumice to get the "flat finish."

20 The outcome of the experiments indicated that it would be desirable to extend the experimental panels to a full size car and, therefore, a proper baking oven was planned that would accommodate one of the largest existing steel passenger cars for the purpose of baking each coat as applied to the exterior and interior surfaces.

21 This oven, as designed and built by the Pennsylvania Railroad Company at its Altoona shops, is 90 ft. 3 in. long, 13 ft. wide and 15 ft. high. The frame work of this structure is made up of 3-in. I-beams for the sides, spaced 5 ft. centers. The roof framing is made of the same sections and curved to conform closely to the contour of the car roof. Each end of the oven has two large doors which can be readily opened and closed for the baking operation. The oven is lined on the inside with ⅛-in. steel plate, and on the outside with galvanized iron of 0.022 gage. The 3-in. space is filled with magnesia lagging, thus effecting the needed insulation. The doors are insulated in a similar manner. Along the walls of the interior of the oven are placed 16 rows of 1½-in. steam pipes, and along the floor, close to the walls, are arranged manifold castings with small lengths of pipe tapped into them at right angles. By this means over 2000 sq. ft. of heating surface is provided. A steam pressure of approximately 100 lb. to the square inch is used, thus making it possible to get an oven temperature of over 250 deg. fahr. Rectangular openings, made adjustable, are provided on the sides near the floor line, allowing the necessary admission of air for circulation. Four 8-in. Globe ventilators are spaced at equal distances in the roof, likewise provided with dampers to regulate the size of the opening. By this means of ventilation, fresh air, which is required for the proper drying of paint, is obtained. as well as providing for the egress of the volatile matter present. Automatic ventilation and steam regulation have not, at the present

time, been applied, but these have been considered advisable, if the result of the experiment seems to warrant a more extended application of the practice.

22 A track is placed on the floor of the oven and connected at each end of the oven with other tracks leading into the regular paint shop where the different coats of paint are aplied to the car before each baking operation.

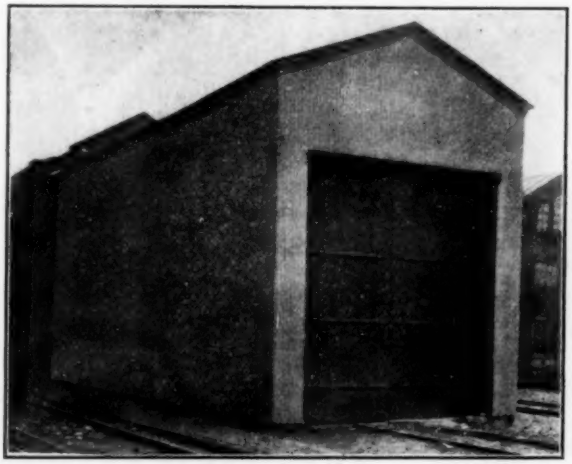


Fig. 1 Exterior Appearance of Oven

23 Photographs of the general appearance of this oven from the outside, and one end of the interior with a car within the oven are shown in Figs. 1 and 2. Fig. 3 shows the steam piping in detail.

### BAKING PAINT ON STEEL

24 The outline of painting a car in this oven is briefly as follows: First, a priming coat is given the exterior and interior of car, which is then moved into the oven and baked for three hours. The temperature at the start is about 160 deg., but rapidly rises at about 1 deg. per min. until a temperature of 250 deg. is reached, requiring about 1½ to 2 hours. The oven is held at this temperature until the lapse of 3 hours, when the car is withdrawn, allowed to cool sufficiently to work upon, after which the surfaces are glazed and depressions and

uneven places puttied. The car then receives its first coat of surfacer, is returned to the oven for 3 hours, baked and removed for additional coats which vary from two to three in number as the needs of the case require.

25 After the last coat of surfacer has been applied and baked, the outside surface of the body of the car is rubbed down with emery

FIG. 2 VIEW OF INTERIOR OF OVEN SHOWING CAR IN PLACE

and oil to produce a flat and smooth surface. The various color coats used, such as tuscan red on the outside, pale green, bronze, and bronze green on the inside, are then put on. Two coats of each color are required to get standard shades. Each coat of color is likewise baked.

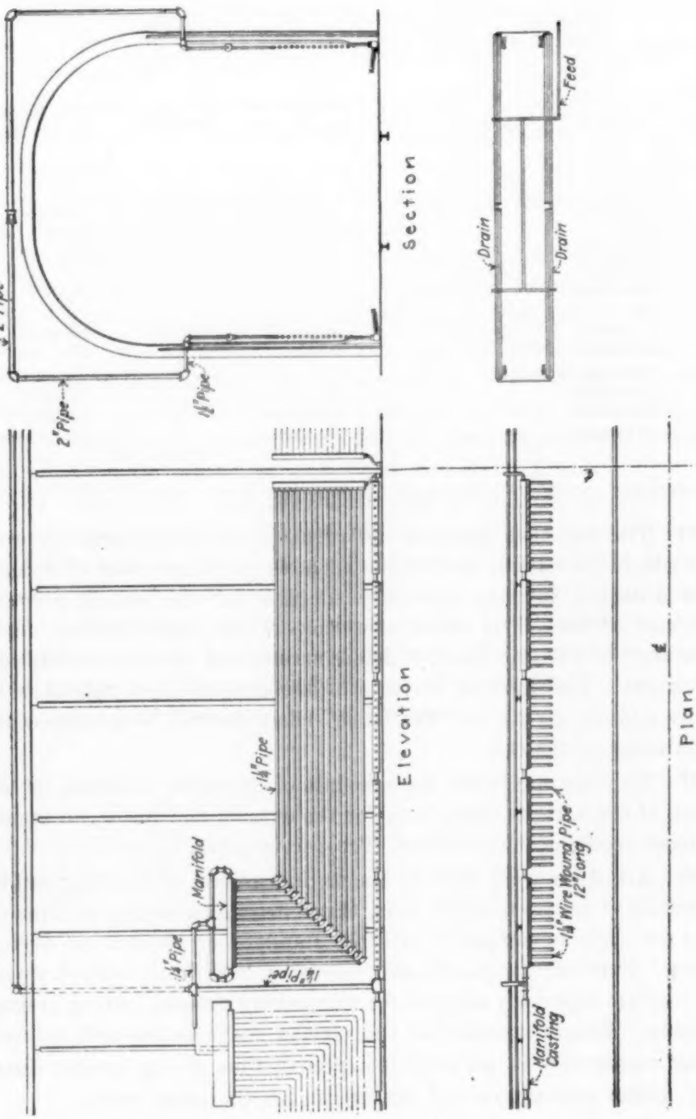



Fig. 3 Detail of Steam Piping

TABLE 1 TIME SCHEDULE FOR PAINTING EXTERIOR AND INTERIOR OF STEEL PASSENGER CARS

| | Outside | | | Inside | | |
|---|---|---|---|---|---|---|
| Period of Work | Body | Roof | Trucks | Body Above Window Sills | Headlining | Body Below Window Sills |
| 1 | 1st prime | 1st prime | .... | 1st prime | 1st prime | 1st prime |
| 2 | glaze | .... | .... | glaze | glaze | glaze |
| 3 | 1st surface | .... | .... | rub-ground | rub | rub |
| 4 | 2d surface | 2d prime | .... | ..... | ..... | ..... |
| 5 | 3d surface | .... | .... | ..... | ..... | ..... |
| 6 | rub | .... | .... | ..... | ..... | ..... |
| 7 | 1st tuscan | 3d prime | .... | 2d ground | 1st green | 1st green |
| 8 | 2d tuscan | .... | .... | stipple | ..... | ..... |
| 9 | stripe and letter | .... | .... | ..... | ..... | ..... |
| 10 | 1st varnish | .... | truck | 1st varnish | ..... | 2d green |
| 11 | 2d varnish | .... | color | 2d varnish | ..... | ..... |
| 12 | 3d varnish | .... | .... | 3d varnish | ..... | 1st varnish |
| 13 | ..... | .... | .... | rub | 2d green air dry | ..... |

26 The car then receives the required lettering, striping, etc., after which the outside and inside surfaces get three coats of a high grade finishing varnish, especially adapted for the baking process. Each coat of varnish is baked at a temperature from 120 deg. fahr. at the start to 150 deg. fahr., which is maintained until the expiration of 3 hours. The interior surfaces of the car are then rubbed with pumice and oil, giving the "flat finish" effect desired, thus completing the painting of the car.

27 To illustrate better the schedule of operation followed, or the timing of the various coats, both for the outside and inside, to secure the most economical conditions, Table 1 is given.

28 All of the work done by the baking process of painting can be accomplished in six to eight days, thus effecting a saving in time of about ten days as compared with the standard or present air drying system. Further, the paints and varnishes have been worked up so that they are especially adapted for this baking process, having greater elasticity. Exact formulae for the various mixtures are well defined, so that uniformity in material is expected, thus giving greater durability, better appearance and longer life for the paint work.

29 The checks and cracking previously found will be considerably lessened, if not almost removed. By oven painting the work is done

under more uniform conditions, which at the present time are so hard to control. It enables the surfaces of the car to be heated uniformly and dried thoroughly, thus removing any objectionable moisture before the first priming coat is applied, which is a very desirable feature of the new method.

30 A considerable saving will be effected by the shorter time that cars will be held out of service when undergoing repairs and repainting in the shops. It is expected that dirt, soot, etc., will not adhere or imbed themselves so readily and that the general appearance of the car will be improved by the baking method.

31 This oven was placed in service the early part of this year and the results of the complete car at this time seem to justify the experiment. They seem to indicate that the results obtained from a small panel can be duplicated in the full size passenger equipment car and that, if this is the case, this method of painting can be used to advantage not only for the painting of steel passenger equipment cars, but for the painting of any other full size steel structure of a similar character where protection and finish are desired.

32 Results and indications at this time seem to justify our expectations that the new process of baking will give, over the present air drying system: (*a*) Longer life of material applied; (*b*) a general appearance as good or better; (*c*) less cost of material at no increase in the labor charge; (*d*) complete sanitation for old cars; (*e*) a considerable saving of time for shopping cars, which results in a saving of shop space. These advantages are offset by the initial cost of installation and operating cost of the oven.

No. 1388 *i*

# PROVISIONS FOR ELECTRIC LIGHTING IN STEEL PASSENGER CARS

By H. A. Currie,[1] New York

Non-Member

Hardly more than perfunctory attention is, as a rule, given to the lighting equipment of a car by the car designer. After all other apparatus and equipment are taken care of, the lighting is considered and fitted as well as possible into the remaining space.

2 From a standpoint of practical consideration of the welfare of passengers, the lighting plays one of the most important parts; therefore, every effort should be made to arrange the light units so that no discomfort be occasioned, and to install the apparatus and wiring so that operating failures be reduced to a minimum. In this connection I might say that the United States postal authorities at Washington are going into this subject very carefully at the present time to insure fair treatment for their postal clerks in the railway mail service; very stringent requirements have been ordered both as regards general illumination and reliable performance.

3 The two essential considerations for the designing engineer to keep in mind in laying out his installation are: (*a*) The arrangement of parts in a manner to allow of easy inspection and repair; (*b*) protection against mechanical injury. Convenience and accessibility of apparatus, fixtures, junction boxes and wiring mean much to the inspector. It is a well-known fact that the average inspector will pay little attention to those parts which are difficult of access, and much better inspection work will result where parts are arranged in an accessible manner. It is of equal importance that the various parts be protected in such a manner as to avoid all possibility of injury to them while the car is in service. The other essential features of the lighting installation are discussed in the following paragraphs:

4 *Axle Generator.* The usual practice is to suspend the generator by swinging links at the inside end of the truck, and belt it to a pulley

[1]Assistant Electrical Engineer, N. Y. C. & H. R. R. R.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

on the axle. For mounting the axle pulley a straight machined seat should be provided in all cases if electric lighting is planned or can be anticipated. Until recent years the universal practice was to provide the regulation tapered axle and allow the manufacturer of electric lighting equipments to adapt his pulley to an unsuitable seat in the best manner he could. The belt and pulley troubles which resulted were disproportionate to any possible advantage from retaining the tapered axles. It is customary for the manufacturer of lighting equipments to provide his own supporting structure adapted as circumstances permit for attachment to the truck. The resulting suspension is at best something of a makeshift.

5 It would be a consummation much to be desired if truck designers would provide a generator support built integral with the truck; the requirements are not difficult and it is certain that the generator builders would be glad to make their machines conform to the truck builder's suspension. As the matter is now handled, nothing causes them more delay and inconvenience than obtaining the numerous details of truck and underframe construction necessary for making an intelligent layout of the generator suspension. In designing the suspension it is desirable that the space required for the belt be kept as clear as possible. The end tie of the truck frame, if used, should not be deep and should be located at a level that will make it possible for the belt to straddle it. Outside brake beams when used are a necessary evil from the standpoint of generator location. Head room for the generator should be considered in laying out deep center girders, brake rigging and piping. All the open space that can be provided about the generator is desirable because it facilitates thorough inspection. The generator terminal board should be attached to the underframe of the car close to the generator and readily accessible.

6 *Battery Box.* On account of the obvious necessity for convenience in handling the heavy batteries, the battery box location has practically been standardized. As the weight and dimensions of elements are almost identical, it is unnecessary to change the hanger design after a satisfactory arrangement has once been used.

7 *Charging Receptacles.* The charging receptacles have been allotted a permanent location on electric lighted equipment. Care should be taken to arrange the wire leading to the receptacles to prevent interference with brake rods, etc.

8 *Switchboard and Regulator Lockers.* (*a*) The switchboard locker should be so located as to be at all times easily accessible to the

trainmen; no pains should be spared in the design and installation of the board; nothing but fireproof material should be used. A receptacle for spare lamps and a report card holder are convenient accessories. (*b*) The regulator locker is generally located under the switchboard and on the generator end of the car. Good ventilation is a necessity. Provision against dampness and dirt is imperative. The regulator lockers should be fitted with locks to guard against accidental or wilful interference with apparatus. In designing lockers for lighting apparatus it is recommended that liberal space be provided so that changing of equipment, repairing, inspecting and testing can be done to the best advantage.

9 *Conduit.* In steel-car construction, metal conduits are almost universally used. In the better type of steel car the interior conduits can be concealed behind metal molding and suitable outlet boxes designed to harmonize with the contour of the molding. Some designers are satisfied to have exposed conduit used exclusively throughout the car. In laying out wiring conduit, direct runs without sharp bends should be used. Care in locating the conduits will facilitate the installation of wires and prevent damage from moisture, etc.

10 *Fixtures.* Where side lighting is used, a satisfactory arrangement can be obtained by designing the fixture to meet the contour of the molding. In center deck lighting, it is advisable wherever possible to arrange the carlines so that a direct support to each fixture may be obtained. On platforms provision for one or two-lamp outlets is sufficient. A plain socket mounted on the platform ceiling has been used in some instances. A better arrangement would be a metallic reflector sunk flush in the ceiling.

11 *Emergency Lights.* It was formerly customary in applying electric light to retain gas lighting as a reserve. Increasing reliability of electric lighting apparatus has made this unnecessary and in the best present practice no gas equipment is provided. For emergencies it is customary to provide holders for candle lamps; but it is only on rare occasions that these have to be used, if the electric equipment is of a good modern type.

No. 1388 *j*

# PROVISION FOR ELECTRICAL EQUIPMENT ON STEEL MOTOR CARS

By F. W. Butt,[1] New York

Non-Member

In providing for the electrical equipment on steel motor cars, several important points should be considered. On account of its metallic construction, the car becomes a negative conductor, or, in other words, the car is grounded, and all electrical apparatus must be well insulated against leakage of the electrical current.

2 Switches, circuit breakers, fuses, etc., should be so located that the arc when opening a circuit will not reach the metal structure of the car. In cases where space is limited, and it becomes necessary to locate circuit breaking apparatus in such a way that there is danger of the arc reaching the metal structure, suitable arc shields of non-conducting and non-inflammable material should be used.

3 Switches, terminals and other apparatus, having exposed live parts, should be protected against accidental contact by enclosing them in boxes or cabinets. This protection is most important where apparatus, such as mentioned above, is located in or near the space which is occupied by passengers.

4 It is sometimes found necessary on account of the restricted space in toilet rooms, motormen's cabs, postal and baggage compartments, etc., to attach electric heaters directly to the sheathing; the heater coils then are necessarily close to the sheathing, and as a means of protection to the paint and varnish thereon, an insulated backing should be applied between the sheathing and the heater.

5 Particular attention should be given to locking bolts, nuts, screws, etc., to prevent them working loose on account of vibration, especially those which are used to secure the apparatus. The vibrations of the motor gearing are transmitted to all parts of the car and they are more pronounced when the motor suspension lug is mounted on the truck transom, without the use of suspension springs. Vibration

[1]Assistant Engineer, Electrical Department, N. Y. C. & H. R. R. R.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

is more easily transmitted through the solid structure of steel cars than it is in cars of wood.

6 In the design of new cars it is sometimes found convenient to locate various members of the structure, especially in the underframe, so the apparatus can be suspended from them without the use of intermediate supports. This is desirable as it is often found that many parts can be omitted from the car. Where heavy apparatus is to be suspended from intermediate supports, large heavy members are required, sometimes complicated in design in order to obtain clearance between parts of the structure or apparatus.

7 Where it is possible, apparatus hangers should rest on the members which support them and not depend entirely upon a vertically bolted or riveted connection. The hangers should be well braced, especially those which hang far below the underframe, to prevent swaying of the apparatus, due to the motion of the car. The hangers can be so designed as to provide the necessary bracing, but to accomplish this odd shapes are often required which increase the cost of manufacture. It is then desirable to provide hangers and separate braces of simple design.

8 When several switches, fuses and other electrical apparatus are required for the motor, control and auxiliary circuits, large switchboard area is necessary, and in some instances, the switchboard has been installed in one of the end bulkheads, occupying most of the space between the body corner and door posts. In recent steel cars, intermediate body end posts are used as part of the general scheme for anti-telescoping provisions at the end of the car. These posts extend from the body-end sill to the body-end plate, and it is recommended, in order to interfere as little as possible with the general anti-telescoping scheme, that two small switchboards be used, one placed in the bulkhead on each side of the body-end door opening, and located as high above the platform as the size of the boards will permit. This arrangement of switchboards provides for the use of short intermediate body-end posts, extending upwards from the body-end sill to the horizontal frame member, located just below each switchboard and connected to the body corner and door posts.

9 In wooden car construction it is necessary to provide ground wires from the various electrical circuits to some part of the car which is a negative conductor. This is unnecessary on cars of steel construction, as the electrical circuits can be grounded at almost any part of the car structure.

10 The steel car is safer than cars of wood construction, as there is no danger of bad fires on account of short circuits. Parts of the structure of a steel car will not become alive, as is sometimes found in cars of wood construction.

11 The wiring conduit on a steel car should be provided for at the time the car is being designed. Unless this is done, difficult bends in the conduit may occur and it is sometimes found necessary to cut and reinforce the structural members.

No. 1388 *k*

# AIR BRAKES FOR HEAVY STEEL PASSENGER CARS

By A. L. Humphrey,[1] Wilmerding, Pa.

Non-Member

Advancement in the development of air brakes has been no less contingent upon the development of rolling stock than the economic handling of traffic through the use of heavier and faster trains is contingent upon the advancements in motive power. A review of the history of railroad transportation development in this country will show a steady and unceasing advance from year to year. Equivalent advancement in the efficiency of appliances such as air brakes was consequently necessary in order that the control and safe handling of longer and heavier trains should not operate as a barrier to these developments.

2 A brief comparison of the conditions existing at the time of the introduction of the air brake with the conditions at present, will show that the advancement in rolling stock has been more rapid than those who have not been in close touch with the situation are likely to realize. For example, the weight on drivers of high-speed passenger engines has increased from 25,000 to 180,000 lb. The drawbar pull of locomotives has increased from 7000 lb. to 30,000 lb.; working steam pressure has increased from 125 lb. to 225 lb.; weights of passenger cars have increased from 20,000 lb. to 150,000 lb. The schedule speeds of passenger trains have increased from 30 miles per hour to 65 miles per hour, and it is not uncommon for speeds to reach as high as 85 to 90 miles per hour.

3 Taking the average weights of trains and average speed at the time the air brake was introduced as compared with the trains and speeds of today, the weight per vehicle has not only increased nearly eight times, but the foot-pounds of energy to be destroyed is nearly 15 times as much. In order to meet the demands of modern service conditions as efficiently as heretofore, means should be provided for dissipating the total energy stored up in this swiftly moving mass in

[1]Vice-President and General Manager, Westinghouse Air Brake Company.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

at least as short a time and distance as before. In fact it is desirable to do this in as much less time as is consistent with comfort to passengers and accuracy of control, in the case of service stops, and in as much shorter distance or time as may be possible in the case of emergency. Not only must the brake be automatic in its operation, but it must be capable at any time and under any conceivable circumstances to produce the maximum possible retarding force within as short a period of time as the known resources available and physical limitations will permit.

4 When we consider that it requires a distance of 8 to 12 miles for a locomotive of modern design, hauling a train of say ten cars, to accelerate to a speed of 80 miles per hour and that this same train should be brought to a standstill within the shortest possible time—or say in one-tenth of the distance required to accelerate to this speed—it is hardly conceivable that this can be done with the means available, which is a retarding force produced by frictional contact of metal shoes against the wheels, which is in turn limited by the adhesion between the wheels and the rail.

5 This factor, viz., the friction obtainable between wheel and rail and shoe and wheel is the basis on which we must start, and upon which we are limited, as to the amount of retarding force obtainable. It is therefore of first importance in designing an air-brake installation to give due consideration to the contact between the wheel and rail and the possible efficiency of the brake shoe. The air brake in itself is practically limitless in the amount of force obtainable, but the practical application of this force is where the line must be drawn. In this connection it is worthy of note that the brake shoe today has about four times as much work to do as it had 30 years ago. The chief effect of this, however, is to destroy the brake shoe at a much more rapid rate, without permitting any material lengthening of stopping distance.

6 The improvements made in air brakes in recent years, which have made it possible to control the present heavy high-speed passenger trains with approximately the same degree of efficiency as the older forms controlled the equipment of their day, have been based on scientific principles and experience in obtaining reliable information and data. The matter of time of transmission of compressed air was not so important a factor with the shorter trains and slower speeds as it is today, where a train running at 80 miles per hour passes over a distance of 117 ft. per second; consequently a few seconds saving in the time of getting the brakes to fully apply is just so much relative gain

in the time and length of stop. With the latest improved pneumatic equipment, the maximum brake cylinder pressure can be obtained throughout a modern train of ten cars in 4 seconds, which is the shortest possible time that this can be obtained by serial quick action through a train of this length. For the purpose of shortening this time serious consideration is being given by some railroad officials to the type of brake equipment used on the New York subway, and known as the "electro-pneumatic," which would not only tend to cut the time of full application in two, but by means of the electric control all brakes are applied simultaneously, which not only assists in shortening the stop but in preventing shocks, etc.

7 Another equally important factor now coming more prominently into use is the application of brake shoes to each side of the wheel, known as clasp brakes. The virtue of clasp brakes, however, is not so much in the aid they afford in shortening the stop as in the equalizing effects of pressure on the wheels, journal box bearings and trucks, the minimizing of lost motion which affects the brakes through increased piston travel, and the less tendency toward wheel sliding while the brakes are applied.

8 While a comparison of the relative merits of a brake equipment, as with most mechanical devices, is frequently based on their maximum capacity, it must be borne in mind that an air-brake equipment must be designed to include flexibility for service operation, in which it is operated 99 per cent of the time and during which time it should be capable of handling smoothly the extreme lengths of trains, while at the same time it must be capable and ready under all conceivable circumstances to produce the maximum permissible braking force in case of an emergency.

9 It is not especially difficult to increase the speed of a train from 30 to 40 miles per hour, but it requires a vastly greater amount of energy to increase the speed from 60 to 70 miles per hour. In like manner, for any given increase in speed, the additional amount of work required of the brakes increases proportionally. If, therefore, the brakes for the heavier trains and higher speeds of today permit of stopping in about the same distance and with the same flexibility of control as could be done with brakes 40 years ago, and with the trains of that period, it is at least gratifying to know that the advancement made in this particular line of railroad development has kept pace as closely as it could consistently with the development in transportation facilities, through which its rate of advancement is largely controlled.

No. 1388

# CAST-STEEL DOUBLE BODY BOLSTERS, PLATFORMS AND END FRAMES FOR STEEL CARS

By C. T. Westlake, St. Louis, Mo.

Member of the Society

Cast steel as applied to underframes and end frames of railroad cars is the result of careful design and painstaking, and thorough development of the art of casting in sand molds. These large steel castings are made in baked molds, confined in massive metal forms, by a special method that assures positively against swelling due to pressure of the inflowing metal, and yet permits yielding to the pressure of the contracting metal when cooling, so that the castings are very accurate in shape and close to size, and are free from shrinkage stresses.

2 Steel is an alloy of iron and carbon and differs from other alloys of iron by being capable of developing all its physical properties to the maximum degree. Its most distinctive properties are rigidity, ability to stand maximum forces without yielding; elasticity, ability to return to normal after being loaded to deflection; ductility, ability to stand distortion beyond its elastic limit without fracture; malleability, permitting it to be forged; tensility, high tensile strength; and weldability, permitting it to be welded by heating and hammering. These properties which steel possesses in a maximum degree distinguish it from all other alloys of iron.

3 Cast steel and rolled steel are produced by the same processes and of the same materials, are of the same chemical composition and have the same physical properties, and cast steel may be substituted for rolled steel, using the same fiber stresses, and its substitution is limited only by the minimum section that can be poured in the molds.

4 As recently as 1893, cast steel was comparatively unknown in car construction, and in that year its introduction began in the use of truck bolsters for freight cars. This was followed a few years later by body bolsters or transoms, and it was only after their use on

---

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

freight cars had demonstrated satisfactorily the reliability of the material and design, that attention was turned to passenger cars.

5 The double body bolster was first to receive consideration for passenger cars, and although, due to casting difficulties, its weight was at first excessive, it was quickly refined and assigned to its proper place with other cast-steel articles. It was found to be so much lighter, stronger and permanantly effective than the built-up type, by forming a one-piece cradle or support for each end of the car body, that its use soon became almost universal in construction of passenger cars.

6 As the demand increased for stronger, safer and less combustible cars, the problem of replacing wood with steel developed many difficulties. The wooden car was the result of many years of experimenting, of cutting and trying with a material easily worked, but as one of the most valuable properties of cast steel is its adaptability to combining a multiplicity of complex parts into a single one of simple form, it was gradually developed from the double body bolster form, first to include end sills, then end and buffing sills; next the end and buffing sills were combined with longitudinal members extending to, and connecting with the double body bolster. Finally these parts, together with many others combined into a single simple member at each end of the car underframe, and comprising so many of the fixed parts that it is now only necessary for the car builder to connect them by center girders and to apply draft and buffing gears and the superstructure to complete the car frame.

7 The ideal underframe should have all connecting members in the same plane so as to avoid buckling due to eccentric leading; it should be so designed that each member will independently perform its individual functions, passing the stresses from one member to the other through the smallest number of properly aligned connections; and all should be so arranged in relation to each other as to form one powerful, compact, shock-absorbing element throughout the length of the car.

8 This can be accomplished to great advantage in cast-steel construction since the metal can be properly distributed in proportion to the stresses. The gusset plates can be placed in the same plane as the flanges of intersecting members, and the whole reduced to minimum weight and to the smallest number of parts with practically no joints. It can be molded to any desired conformation, can be shaped to any curve, useful or ornate, without the use of expensive dies, can be provided with necessary projections joined to the main members by

proper fillets. Openings may be provided with finished and reinforced edges, and all parts may be molded to symmetrical, pleasing contour, all edges rounded and a complete, practical, operative device, emanating from a single source furnished to the car builders ready for application.

9 As the rounding of curves necessitates the use of convex ends to the car body, the central portion of the ends is most exposed and liable to receive initial impacts, and this portion should be made strongest and most capable of properly transmitting the force of impacts to the balance of frame.

10 The underframe receives the force of end collision as a column load on its longitudinal members, while the end frame receives it as a transverse load on exposed members supported at their ends. As it is impracticable under these conditions to make the end frame equally as strong as the underframe, provision should be made for protecting the end frame against destructive forces. The underframe should be arranged so as to receive the initial impact, and if the encountered force is sufficient to destroy it, it should fail in such manner as to form additional protection to the end frame.

11 This is accomplished in cast-steel construction by arranging the parts of the longitudinal members so that when loaded to destruction by a collision force, the end portions yield upwardly, thus folding the exposed portion of the platform up against the end of the car body, and forming an addition to the end frame to assist in distributing the force to all the longitudinal members of the superstructure. The advantage of this construction has been demonstrated in wrecks when this identical action has taken place, the safety of passengers assured, and the property loss kept low.

12 The cast-steel platform as now provided for blind end cars, comprises the buffing sill having recesses for the buffer foot plates, holes and brackets for the buffer stems, pockets for the buffing device, brackets for safety chains, lugs for draft gear, brackets for drawbar carry irons, anti-telescoping plate, extensions of the center sills and bottom chords of the side sills, all of the double body bolster members including side bearing arches and extending for a distance of over 14 ft. inward from the end of the car to a point considerably back of the truck center, and counting rivets, gusset plates and connecting angles, combining more than 1000 pieces into a single, powerful, shock-absorbing element of less weight than fabricated material of the same strength.

13 The cast-steel platform and double body bolster for vestibule cars comprises all the parts enumerated for blind end cars, and in addition, includes the exposed platform longitudinal members, step risers and end sill, measures over 17 ft. in length, is made of a single piece, and is also of less weight than fabricated material of the same strength.

14 Since the government has taken a hand in the construction of cars used in its service, stronger body end frames are being used, and as the end of the car is the first to encounter end collision forces, it reasonably deserves closer and more careful consideration.

15 Most damage is produced by end collisions and to protect life and property from them, the colliding object must be prevented from entering the car. To accomplish this, the end frame and end portion of underframe should be constructed so as to distribute the force of collision into all the longitudinal members of the car, passing it into the largest mass, utilizing every particle of available inertia to absorb the force without permitting it to reach and act upon the contents or occupants of cars.

16 The end frame proper should be designed so that when a single member is loaded, all will act with it, and this can be accomplished only by connecting them so as to form a single mass, and best by forming them in a single piece as in cast-steel construction.

17 In designing the cast-steel end frame we assume it to be a beam supported at its upper and lower ends and loaded at a point about 18 in. above its lower end. We provide connections between the end frame and balance of car frame of sufficient value to develop the full transverse strength of the end frame; the vertical members of end frame are connected by horizontal members so that in case the end frame is loaded to destruction the connections are sufficient to disrupt all the longitudinal members of the car frame, and when they yield all parts will be forced toward the center of the end of the car and tend to prevent one car telescoping the other.

18 Cast-steel parts weigh less than built-up members carrying the same load since the metal in castings can be properly distributed in proportion to stresses. In built-up construction the metal overlaps at the joints and this, together with the rivet heads, makes an additional weight which in cast construction is avoided. In the latter, reliance is placed in a single solid member and, as there are no joints, there is no chance of their being imperfect or becoming loose.

19 The advantage in cast steel to the car builder is also very

great. To produce a platform of the built-up type at least eight different classes of material are required. This comes from eight different manufacturers, frequently from as many different points of production, much of it in less than car load lots, and all has to be requisitioned, purchased, received, stored and recorded for use on each particular lot, and in order to reduce storage space and avoid congestion in the car plant, all deliveries have to be carefully and accurately timed, and followed up. Then each material has to be passed through the different departments of the car plant to be cut, shaped, punched, drilled and the same timing and tracing methods used, so as to have all parts completed at the proper time. When cast steel is used but one material is purchased from a single plant, only one piece is handled, that in car load lots, and when it arrives it is immediately ready and available for application without storage or re-handling, facilitating completion of the car by leaving more car plant machinery available for other work.

20 A plant capable of producing castings of this nature in quantities to meet requirements of the many car plants must have buildings of extensive area and equipment in proportion, as it ordinarily requires about ten days for a casting to pass through the various processes of casting, cooling, roughing, cleaning and machining, and an accumulation of ten days' output has to be constantly accommodated. All handling and conveying apparatus must be in duplicate so as to insure uninterrupted operation and machines for finishing must be of the highest grade and maintained in perfect condition to produce accurate and proper results.

21 In car construction cast steel stands preëminent as the best material for reducing to the minimum the weight and number of parts while maintaining requisite strength and other essential properties, and its popularity and use will proportionately increase as its benefits and advantages become more generally recognized.

No. 1388 *m*

# SPECIAL ENDS FOR STEEL PASSENGER CARS

By H. M. Estabrook, Dayton, O.

Member of the Society

After the passenger car had emerged from the stage-coach type of construction the box-like shape of car was introduced with straight longitudinal floor sills and with straight vertical side and end posts. These members were of wood, the ends of the longitudinal floor sills being tenoned into mortises in the wooden end sills. The vertical side and end posts were in like manner tenoned into the side and end sills at their lower end and likewise into the wooden side and end plates at their upper ends. These side and end posts were maintained in their several positions, by wooden spacing blocks or bridging, and the whole structure tied together by means of iron rods and bolts. These spacing blocks served further the double purpose of affording a foundation for securing the outside panels and the wooden interior finish.

2 Several types of roof were quite prevalent in early passenger car days, among them being the round top or omnibus roof, which has again made its appearance in steel passenger cars in some parts of our country. Another type of roof was the Ogee, or turtle-back, and later came the monitor, or raised deck roof. The prevailing type of hood projection over the platforms was the "duck's bill" type, as illustrated in Fig. 1, which also furnishes a good idea of the framing employed in those days. Fig. 2 shows end framing of these same cars.

3 A little later the projecting platform hood was changed from the "duck's bill" type to the bull-nose type. Figs. 3 and 4 show respectively a longitudinal section and exterior of these cars. Fig. 5 shows the end construction and Fig. 6 the standard framing employed in the first bull-nose hood cars in the early eighties. Up to the middle eighties no systematic attempt had been made to strengthen the ends of cars. The platform members were all of wood and the end framing of the car had not experienced much change in the way of strength-

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

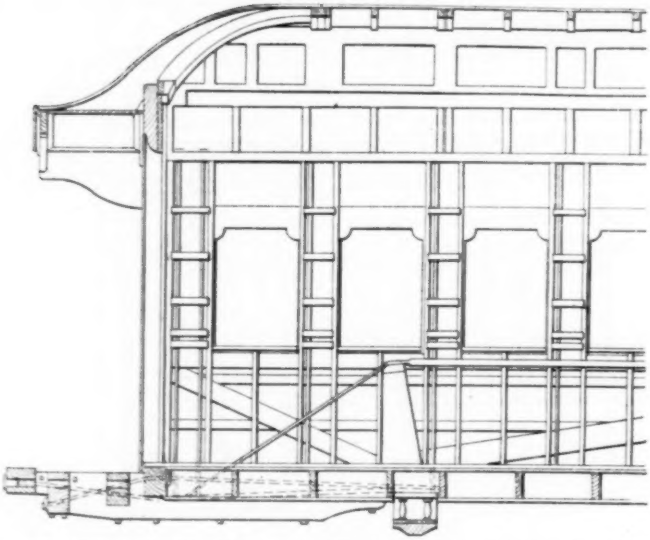


FIG. 1 "DUCK'S BILL" TYPE OF HOOD PROJECTION OVER PLATFORMS

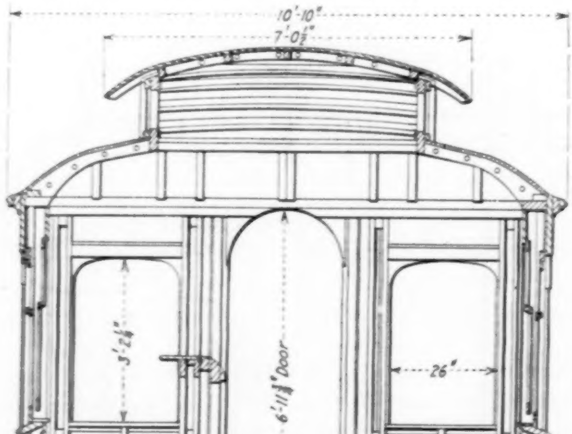



FIG. 2 END FRAMING FOR "DUCK'S BILL" HOOD PROJECTION

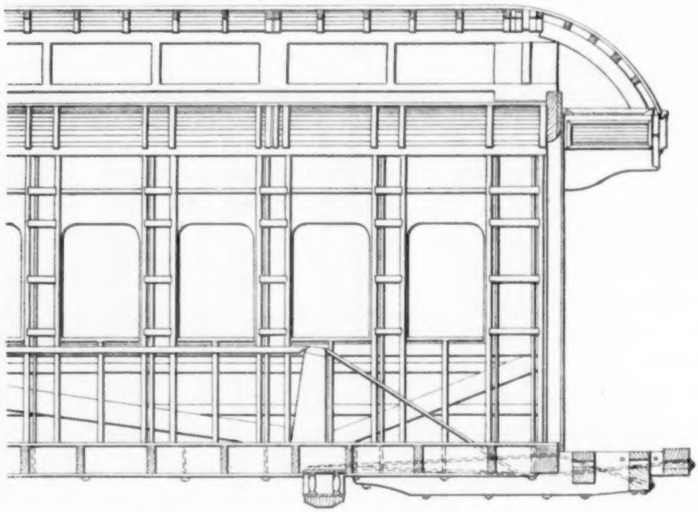


FIG. 3 SECTION OF BULL-NOSE TYPE OF CAR

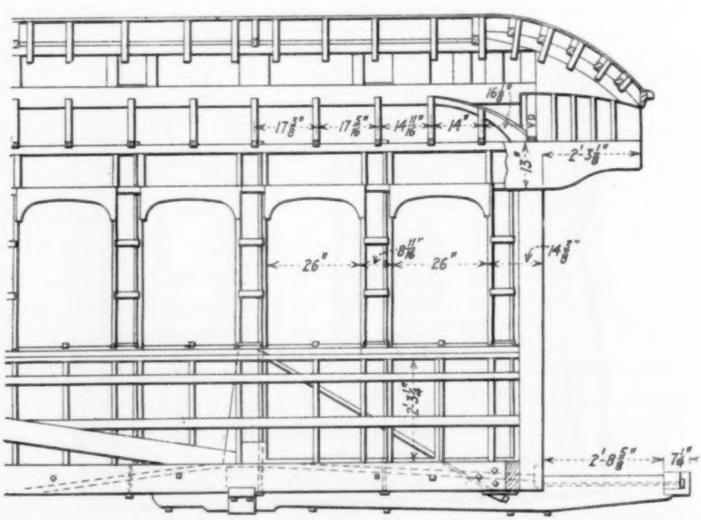



FIG. 4 BULL-NOSE TYPE OF CAR, FRAMING EXPOSED

ening from the earlier types. With the advent of the narrow vestibule in 1887, which was immediately followed by the broad vestibule in 1888, came the demand for a stronger end.

4 About the year 1890 there was brought into use what was known as an "anti-telescoping" end framing. This construction consisted of double side and end sills with a steel plate 8 in. by ½ in. from 18 to 24 ft. long, sandwiched into the double side sill, with the end of these plates turned so as to form a foot against the end sill. The double end sill had a steel plate 8 in. by ¾ in. and the length the width of the car, sandwiched into the end sill. The end posts of

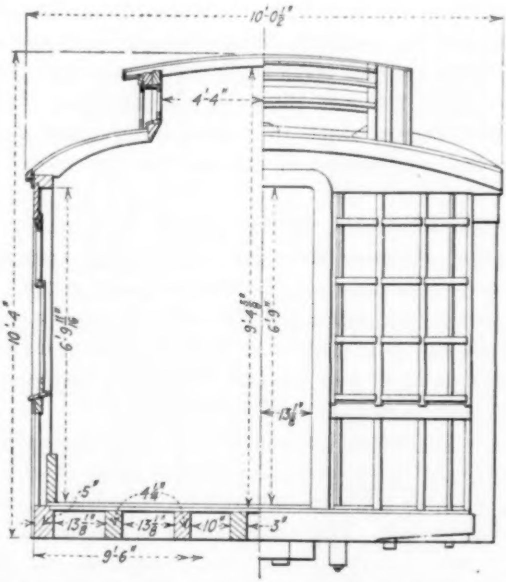



Fig. 5 End Construction of Bull-Nose Type of Car

the car were reinforced by steel bars 3½ in. by ¾ in., extending downward through and bolted through the sandwiched end sill and having their upper ends extending upward and bearing on and bolted through a steel plate 6 in. by ⅜ in., which was bolted to the oak end plate of the car. This stiffening plate extended across the width of the car and the ends of the steel plate being turned so as to form a foot upon the side plate of the structure. This anti-telescoping construction is illustrated in Fig. 7. This design of end framing came into general use throughout the country and is in use today in the

majority of wooden passenger cars built since 1890. It is interesting to note that this anti-telescoping framing is the same, with some modifications and additions, as was adopted by the United States Government for the construction of full postal cars and known as Specification No. 1.

5 Somewhat later this type of end framing was elaborated upon by the use of a heavy steel angle flitched into the end sill, with the end still further reinforced by a 20 in. by ½ in. steel gusset plate on the under side of the sills, and by the use of steel Z-bars in the end posts and a heavy steel angle introduced into the construction of the end plate of the car.

6 The increased weight of the vestibules and anti-telescoping end framing developed the necessity for a stronger platform construction than the old style wooden platform member that had been used for many years. About the year 1895 the standard steel platform, composed of steel I-beams, came into general use, and was employed continuously until the advent of the steel car superseded it by other designs.

7 Notwithstanding the frantic efforts of Congress toward the general adoption of steel passenger cars, it has been stated upon reliable authority that no vestibuled wooden passenger car, in the construction of which was employed the anti-telescoping end framing, in a straight-on end to end collision (although frequently having the ends concaved) has ever had the end crushed in to the extent of the adjoining car body telescoping and entering it.

8 The United States Government in seeking to strengthen the end construction of postal cars adopted this form of anti-telescoping end framing with the addition of two 7-in., 23.46-lb. steel bulb beams on either end of the car. These bulb beams have their flat base resting against the outside of the reinforced end posts of the car, being located in line with and immediately behind the vestibule diaphragms and face plate. At its lower end, this bulb beam has the head and web notched out with the base flange extending downward through the flitched end sill, the main body of the beam resting upon the 1 in. thick steel plate on top of the buffer beam. At the upper end these bulb beams have the web and bulb head sheared diagonally so the base flange extends upward on the outside of the end plate of the car framing, and through this flange passes the top piston stems of the vestibule mechanism. This type of construction is now obsolete in postal cars, Congress having enacted a law requiring them to be of steel construction.

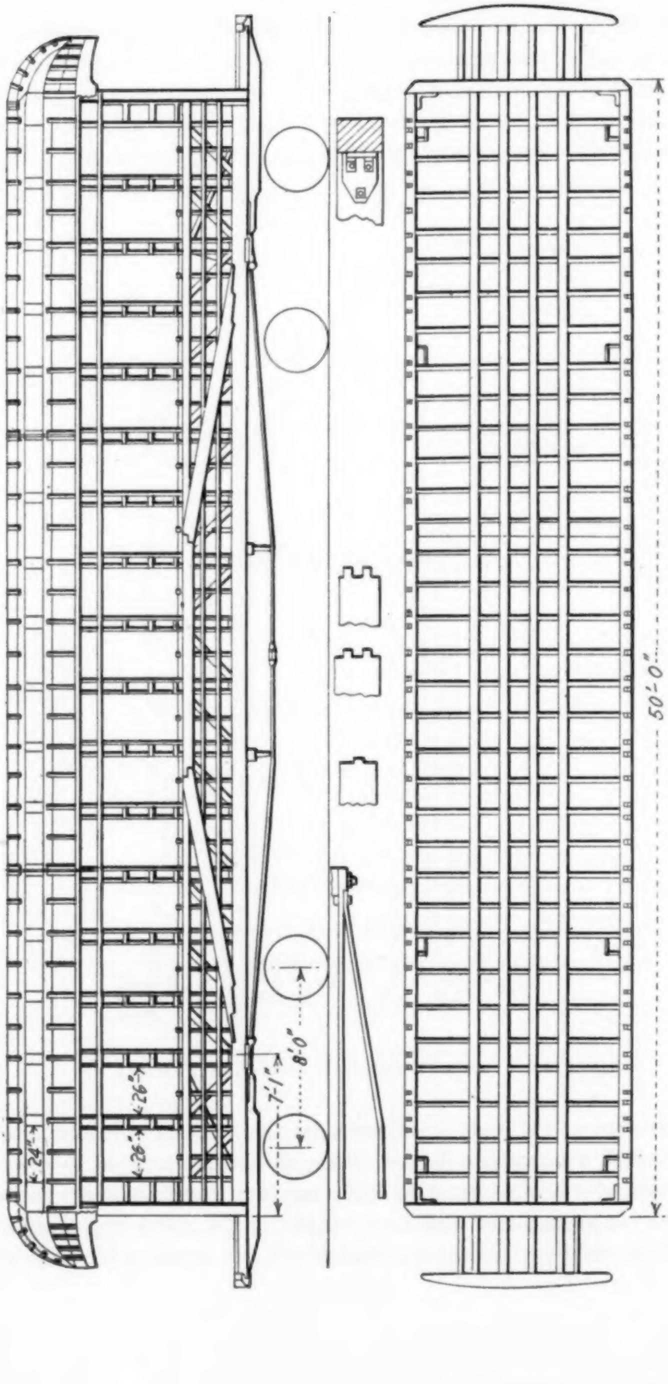



FIG. 6 STANDARD FRAMING EMPLOYED IN FIRST BULL-NOSE CARS

9 When the steel passenger car made its appearance about the year 1905, the passenger car entered a period of transition and evolution from which it has not yet entirely emerged with a recognized standard form of construction. The wooden car had attained a degree of uniformity that established it as an accepted standard. In the

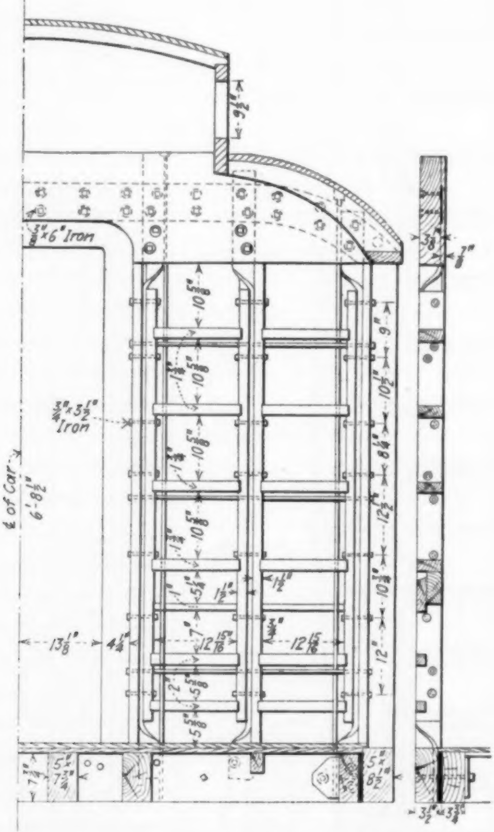



FIG. 7 ANTI-TELESCOPING IRON END FRAMING

construction of the early steel passenger cars, as was probably natural, an attempt was made to follow closely the lines employed in the construction of wooden cars, with the result that the first steel cars were inferior in strength of end construction to the prevailing wood construction, but the evolution has been rapid, one improvement following

close upon the heels of another. In the entire history of car building, there has probably not been devoted so much concentrated thought and study to the improvement in design, by the most expert engineering talent of the railroads and car builders, as has been shown since the introduction of steel cars. This has resulted in rapid improvement of end construction until we have today reached a design

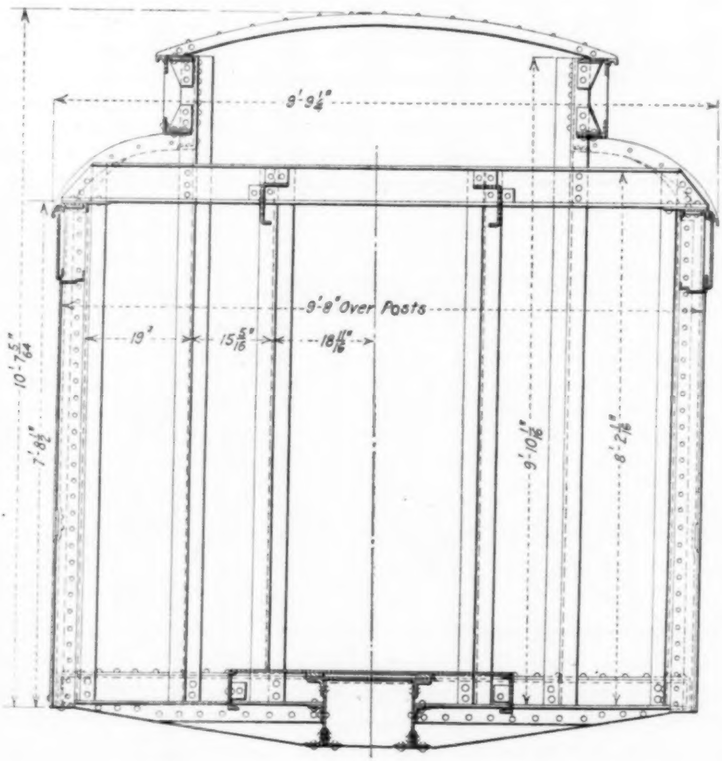



FIG. 8 BODY END FRAMING, TYPE SHOWN IN FIG. 9

that is considered practically standard. This development has no doubt been hastened by the action of Congress relative to steel postal cars and the coöperation of committees of the railway mail service, the railroads and the car builders, to the end that a standard specification for the strength of the various parts of the car, and especially

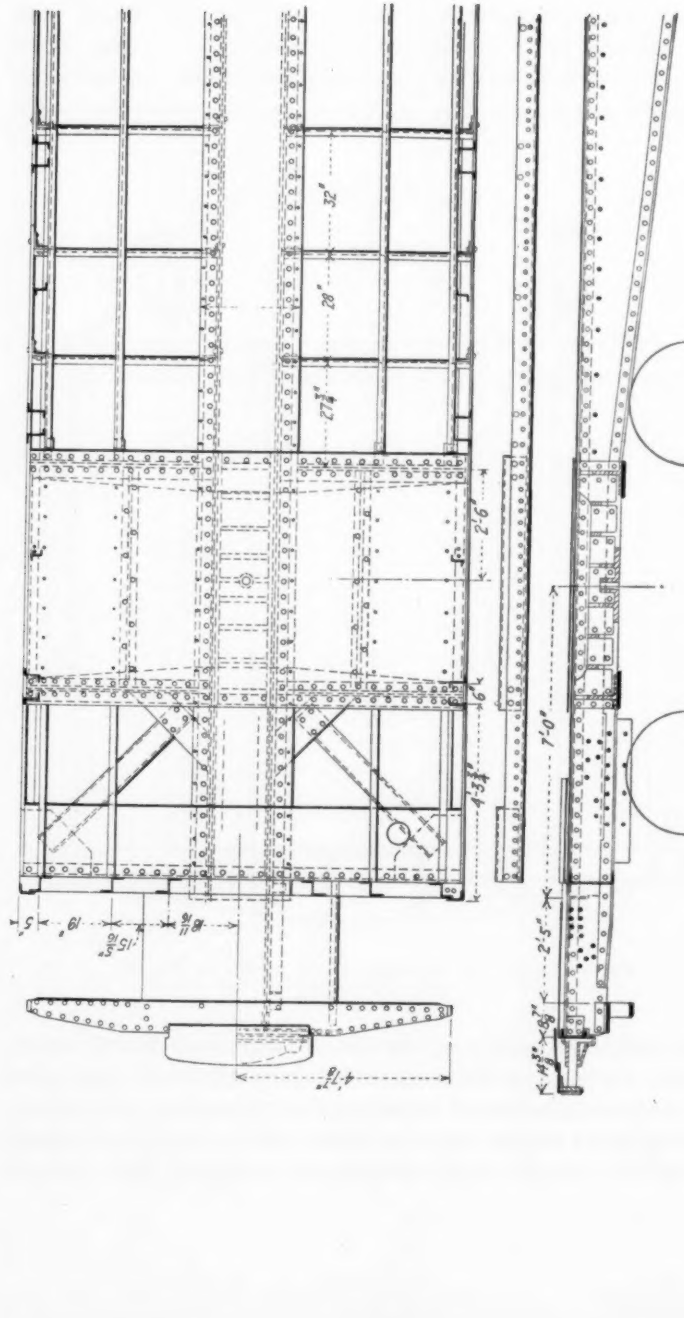



FIG. 9 STEEL CAR CONSTRUCTION, CENTER SILLS EXTENDING FULL LENGTH OF CAR

the end construction, has been adopted by the Postoffice Department in which it is provided that:

**The maximum end shock due to buffing shall be assumed as a static load of 400,000 lb. applied horizontally at the resultant line of forces acting as the center line of the buffing mechanism and at the center line of draft gear, respectively, and shall be assumed to be resisted by all continuous longitudinal underframe members below the floor level, provided such members are sufficiently tied together to act in unison.**

**The sum of the section moduli of all vertical end members at each end shall be not less than 65 and the section moduli of the main members, either forming or adjacent to the door posts, shall be not less than 75 per cent. of this amount. The horizontal reactions of all vertical end members at top and bottom shall be calculated from an assumed external horizontal force, applied 18 in. above the floor line, to all vertical members in the proportions given, such force being of sufficient amount to cause bending of all vertical members acting together, and top and bottom connections of vertical members shall be designed for these reactions. Except where vertical end members shall bear directly against or be attached directly to longitudinal members at either top or bottom, the assumed reactions shall be considered as loads applied to whatever construction is used at end sill or end plate and both these last named members shall have section moduli, respectively, sufficient to prevent their failure horizontally before that of the vertical end members. All parts of the car framing shall be so proportioned that the sum of the maximum unit stresses to which any member is subject shall not exceed the following amounts in pounds per square inch—these stresses, unless otherwise stated below, are for steel having an ultimate tensile strength of from 55,000 to 65,000 lb. per sq. in.:**

***Bolsters of Rolled Steel*—Stress shall not exceed 12,500 lb. per sq. in.**

***Sills and Framing of Rolled Steel*—Stress shall not exceed 16,000 lb. per sq. in.**

**When cast steel is used the allowable stresses may be the same as for rolled steel except tension stresses, which must be at least 20 per cent less than those allowed for rolled steel, as specified above.**

10 To meet these requirements, there are at this time three distinct forms of construction employed: The one most generally employed is illustrated in Figs. 8 and 9, which is composed of rolled-steel sections with the center sills running the full length of the car from buffer beam to buffer beam. Another type is that in which the rolled steel center sills connect at the bolster with a steel casting, forming a combined body bolster, center and side sills, and end sills, as illustrated in Figs. 10 and 11. Another type is that in which the rolled-steel center sills connect at the bolster with a steel casting, forming a combined body bolster, center and side sills, end sill and the entire end frame of the car, as illustrated in Fig. 12.

11 In the first form of construction, shown in Figs. 8 and 9, rolled sections are employed entirely. The members forming the

center sill construction extend the full length of the car from one buffer beam to the other and all other longitudinal members, such as side sills, belt rail, etc., extending the full length of the car body and in the case of vestibuled cars, the rolled section side plate extends the full length of the car from one vestibule corner post to another. The end sill is usually composed of pressed or rolled shapes riveted to the center-sill construction and extending laterally outwards to the sides of the car, the ends of the side-sill members butting against and being riveted to these end-sill members. The upper end plate of the car is composed of rolled or pressed sections extending in one piece across the width of the car and attached to the longitudinal side plates by connecting angles and gussets. To this end plate are also attached the longitudinal members of the upper deck sides. The end posts are rolled or pressed sections, usually Z-sections, extending downward to the bottom line of and riveted to the end sill. The upper ends of these posts extend upwards to the top line of and are riveted to the end plate. The nose piece or buffer beam is composed of rolled channels with their flanges turned inwardly towards each other, presenting their smooth surfaces on the outside, these channels being formed to suit the contour requirements of the vestibule, the channel members forming a box construction with top cover plates.

12 This buffer beam extends across and is riveted to the outward ends of the center-sill construction, from which it will be observed that the purpose of this design is to transmit the end buffing shock to the center-sill construction. The vestibule corner posts are rolled channel or Z-sections, whose bottom ends extend down into and are riveted to the outer ends of the buffer beams and whose upper ends are riveted to the vestibule end plate and to the upper longitudinal side plate of the car body, which projects beyond the end of the car body to meet and to connect with this vestibule corner post. The center vestibule posts are 6-in. I-beams whose lower ends extend downward through and are connected to the buffer beam member and whose upper ends extend upward to and are connected to the vestibule end plate steel angle. Between the upper ends of these center vestibule posts and the end of the car body, are longitudinal compression members in the form of steel channels or angles. These rolled section corner posts, door posts and vestibule door and corner posts, are encased in light steel casings formed to give them the finished appearance of the same members in a wooden car.

13 In stub-end cars of this type of construction, the buffer beam

is of considerably heavier construction than in the vestibule car, and is usually composed of a built-up box construction or a one-piece steel casting, this buffer beam being secured immediately to the outside face of the end sill. In this construction there is usually employed a

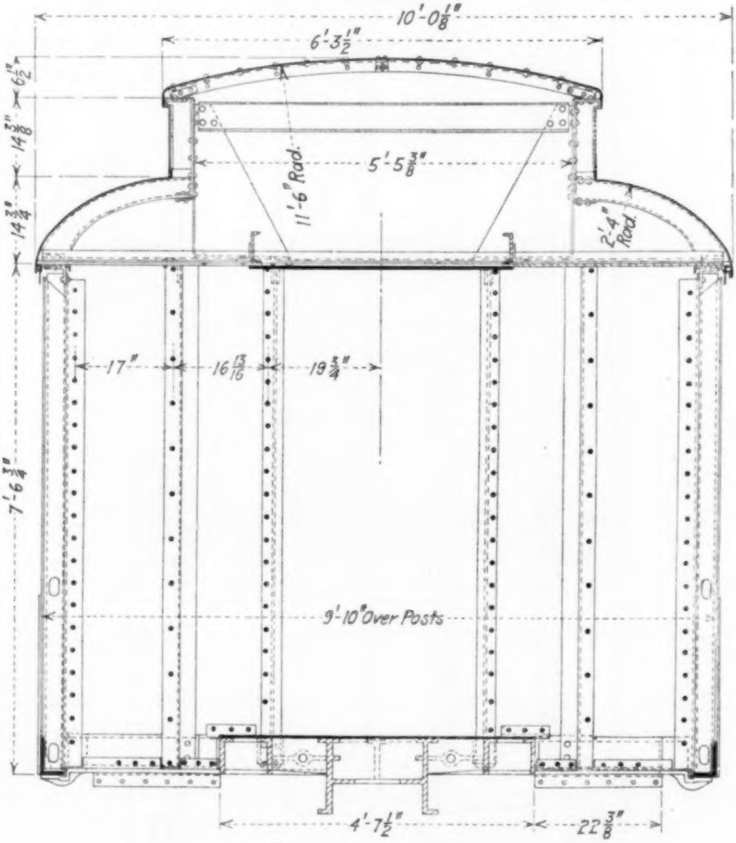



FIG. 10 BODY END FRAMING, TYPE SHOWN IN FIG. 11

much heavier vestibule center post than in the vestibuled car. These vestibule posts, usually being a 12-in. I-beam, are located immediately in line with and behind the vestibule diaphragm and face plate. The end-post construction is much the same as described for the vestibuled car, there being a difference, however, in the construction of the end plate, which in the stub-end car is a pressed channel section formed

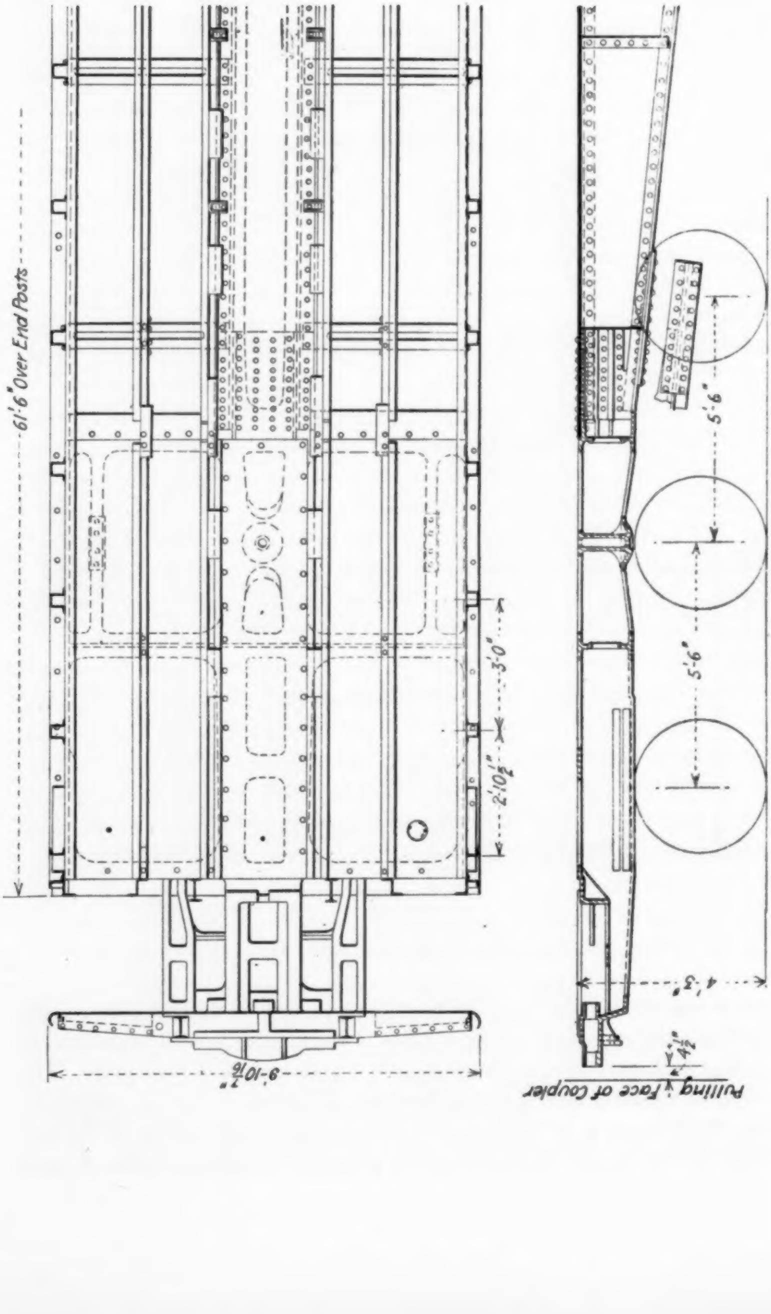



FIG. 11 STEEL CAR TYPE, CENTER SILLS CONNECT AT BOLSTER WITH A STEEL CASTING

to suit the contour of the car end, this channel end plate being placed across the end of the car in a horizontal plane, and into and riveted to this channel end plate are the upper ends of the corner posts, end posts and vestibule posts.

14 In the second type of construction referred to, a steel casting is employed forming the body bolster and platform to which the center-sill construction is riveted to this steel bolster. This construction is illustrated in Figs. 10 and 11, from which it will be observed that the center sill construction, the end sill, platform and buffer beam are all embodied in one steel casting. The end-post construction, the corner posts, vestibule corner and center posts are practically of the same

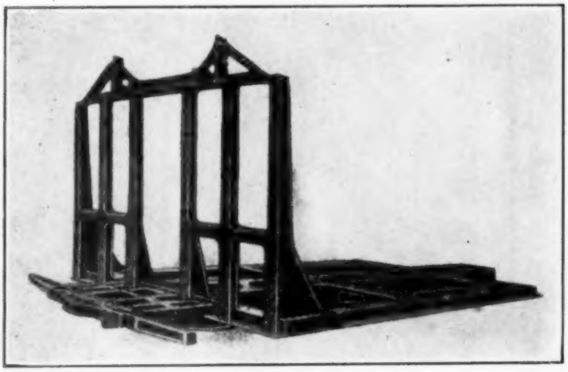


Fig. 12 Integral Steel Casting Used in End Frame Construction

construction as described for the built-up type, the difference being in the method of attaching the lower ends of these posts. The steel castings have openings or pockets in the end sill and buffer beam members, in which the lower ends of these posts rest and are riveted to the casting. The construction of the end of the car body, the vestibule and hood are substantially as described for the built-up construction.

15 This type of construction for the stub-end car is substantially the same as that just described, the exception being that the steel bolster and end-sill casting takes the place of the built-up type of center and end-sill construction, the end post, corner post and upper end construction being identical in the two types.

16 In the third type of construction referred to the entire bottom framework of the car from the bolster outward to the platform and

buffer beam, is one integral steel casting, and the entire end framing of the car is one integral steel casting, as illustrated by Fig. 12.

17 In referring to the three types of construction just outlined,

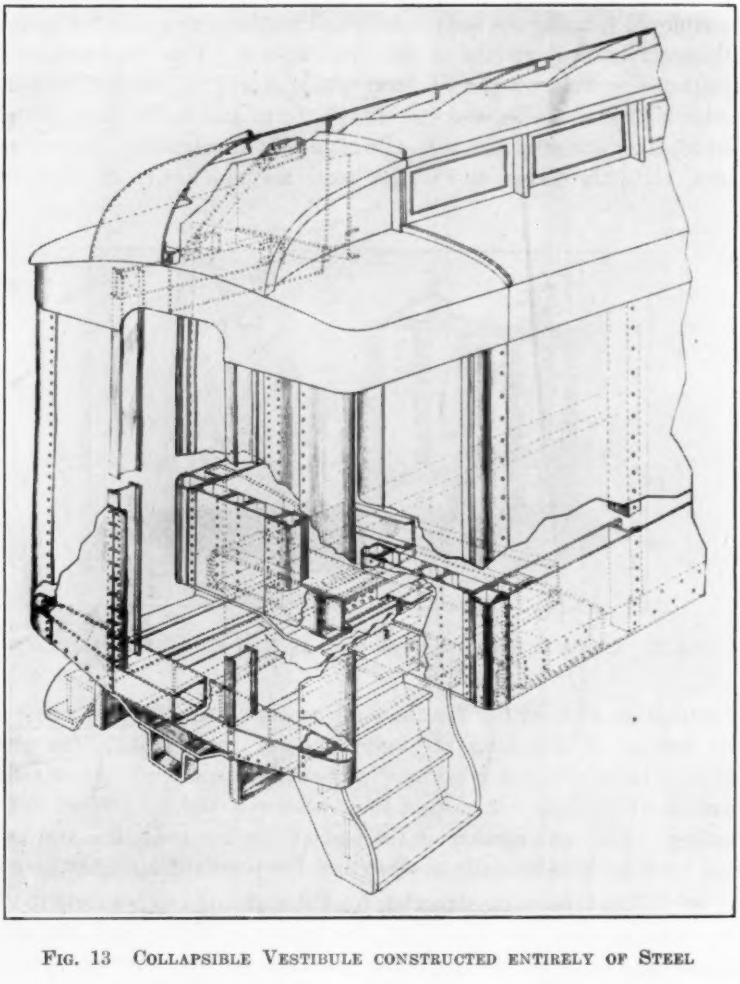


FIG. 13 COLLAPSIBLE VESTIBULE CONSTRUCTED ENTIRELY OF STEEL

it must be understood that reference is made to them only as types, and no attempt is made to describe the construction of any one railroad or car builder in particular, or to undertake to establish any of

the forms described as being a standard, as the details of construction vary to a considerable degree with different railroads and builders.

18 It is of course apparent that the weight of the steel car is

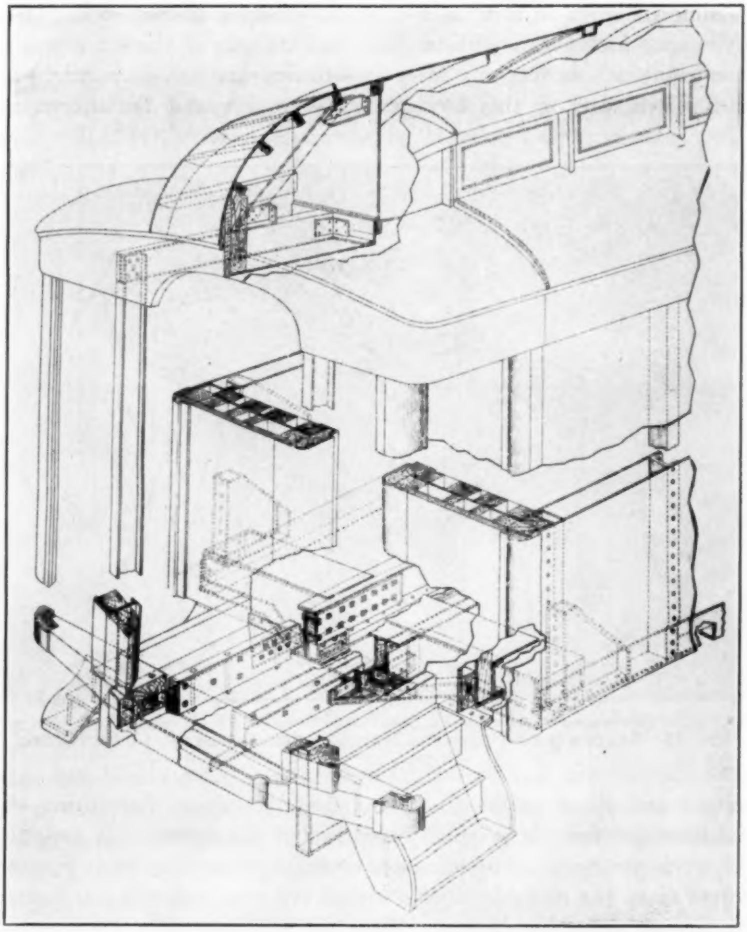


FIG. 14 COLLAPSIBLE VESTIBULE MADE OF A SERIES OF WOODEN POSTS TO SECURE ADVANTAGE OF ELASTIC AND CUSHIONING PROPERTIES OF WOOD

much greater than a car of the same size of wooden construction, and that the wooden car possesses in itself a natural elasticity to absorb

buffing shocks such as are produced by collision that the steel car does not furnish. Hence, in the development of the steel car, with the enormous increase in weight of trains and the high speed at which they run, there has been a growing tendency to increase the strength of the structure with the view of making it as nearly indestructible as possible in order to compensate for the absence of elasticity. It is also apparent that, notwithstanding the strength of the structure, if it encountered an opposing force of sufficient magnitude, it might be annihilated, and so this strengthening process, and the increasing

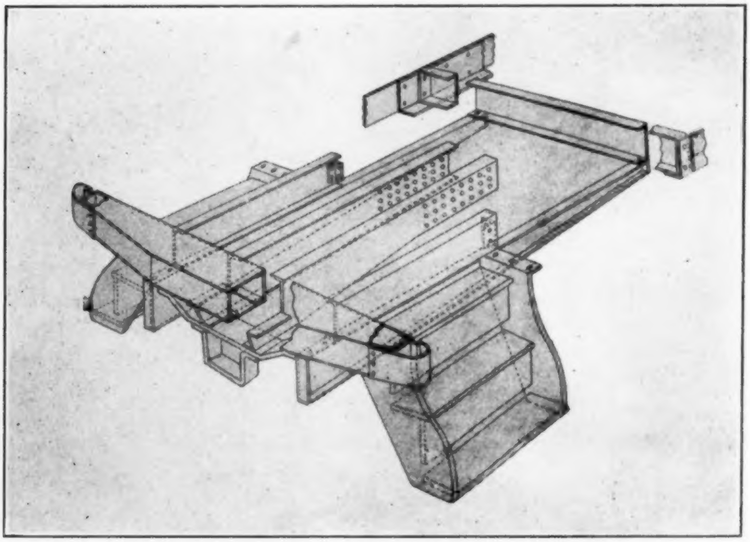


FIG. 15 SKELETON OF PLATFORM MEMBERS FOR ALL-STEEL CONSTRUCTION

weight and speed might go on indefinitely without furnishing the result sought for. It is equally true that if the structure is designed for such strength as to be indestructible, when the two opposing forces meet, the movable objects within the cars, which is the human load, must suffer the damage. To avoid this possibility the idea has been evolved to construct that portion of the end of the car between the end of the main body and the vestibule face plates, these members being all such parts as are embraced in the platform, vestibule and hood covering the vestibule, so that it will collapse under a less shock than would be required to crush in the end of the car body itself.

19 This idea is based on the theory that in a train in which there are say ten vestibuled cars, there is the space between the main bodies of each two coupled cars occupied by the platforms and vestibules of approximately 8 ft., or in a ten-car train a space of approximately 80 ft., of shock absorbing space, which, if properly utilized in the instant of collision, would remove to a large degree the shock and resultant damage to the car body itself and likewise lessen the possibility of damage to the persons of the passengers. From this idea has developed what is termed a collapsible vestibule. It is generaly conceded that if two vestibuled cars coupled together could maintain their

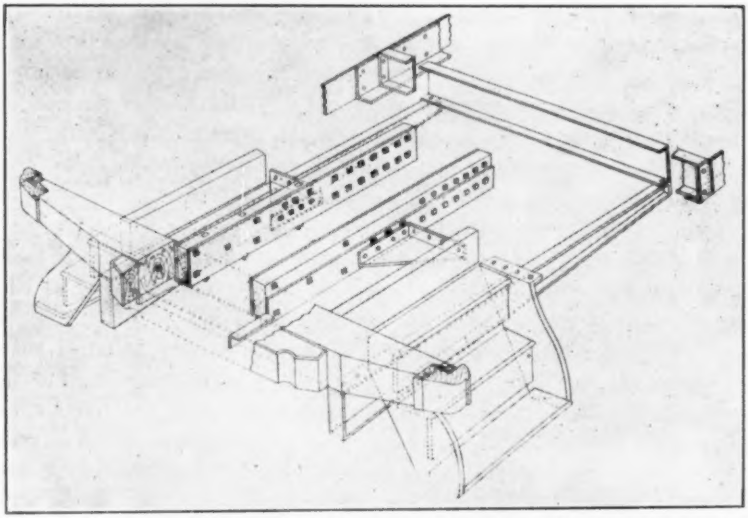


FIG. 16 SKELETON OF PLATFORM MEMBERS, STEEL AND WOOD CONSTRUCTION

respective horizontal planes at the instant of shock due to collision, there could be no telescoping and that telescoping is due to one car assuming, at the instant of collision, a higher or lower horizontal plane than its adjoining neighbor, causing one to ride the other with the resultant telescoping effects.

20 It is generally conceded, that in cases of two cars tending to telescope, the point of maximum shock is never over 20 in. above the floor line. In the Government postal car specifications, this point has been definitely fixed at 18 in. above the floor line, and with this in view the end posts are reinforced for a distance of about 4 ft. above the floor line by steel angles riveted to the Z-bar end posts.

21 A general idea of this collapsible vestibule is afforded by Figs. 13 and 14. Fig. 13 shows the construction entirely of steel, while Fig. 14 shows a series of wooden posts and platform and vestibule members in addition to the steel members to secure the recognized advantage of the elastic and cushioning properties of the wood.

22 In this construction the longitudinal sills and floor members are designed to stop at the end sill of the car body proper, the end of which is sheathed with a heavy steel plate extending in one piece vertically from the roof downward to the bottom of the end sill. If the shock of collision is not entirely absorbed by the vestibule members before the end of the car body proper can be crushed, this plate will tend to pull the roof downward and cause the direction of the oncoming car to deflect obliquely upwards instead of the two cars telescoping. Further to offset the effect, should the two cars change their horizontal planes in collision, pressed steel shapes in the nature of anti-climbers are placed below the buffer beam and platform.

23 Fig. 15 shows the skeleton of the platform members for the all-steel construction, and Fig. 16 shows the skeleton of the platform members where wooden features are employed.

24 The vestibule diaphragm posts are constructed of heavy steel I-beams rigidly secured at the bottom to the buffer beam and at the top to the vestibule end plate and longitudinal braces.

25 The platform, vestibule and hood members are designed with a view to withstanding all shocks incident to regular service, but in abnormal shocks, such as would result from collision, the rivets connecting the various members would shear off with the exertion of less energy than would be required to crush the end of the car body, thereby causing the vestibule to collapse, absorbing the shock and furnishing a cushion between the two car bodies proper. It is assumed that in case of a collision these would be the only parts seriously damaged, and the car could be repaired and replaced in service with a minimum of expense and delay.

26 The entire collapsible vestibule, comprising the platform, vestibule and hood, is constructed as a unit, detachable and separate from the car body proper and can be applied after the car is built or in the alteration of cars already built and is equally applicable to cars of either steel or wood construction.

27 The object of the collapsible vestibule is, first, to protect the lives of the passengers and secondly to protect the body proper of the car from serious damage.

## No. 1388 *n*

# DISCUSSION ON STEEL PASSENGER CAR DESIGN

George Gibbs. On this occasion when the subject of steel passenger car design is under discussion, it may be of interest to make a brief reference to the early history of this important innovation in railway operation which had its origin in connection with the provision of car equipment for the first rapid transit subway in the City of New York. The writer was at that time consulting engineer of the subway construction company in charge of car design and construction. It was obvious that the exacting requirements of the contemplated service, which involved tunnel operation at a high schedule speed with closely spaced trains crowded with passengers, must be conducted with all possible precaution against accident and, further, in a way such as to minimize the fatal consequences of any accident which might occur in spite of such precautions. The two consequences most to be feared from an accident are the breaking up of the cars in the train and the setting of a fire in the wreckage; on an open railway line the consequences of these are serious enough, but in a subway or tunnel they are potentially much worse, because of the confined space which prevents the prompt escape of passengers from the wreckage.

Cars for such service, therefore, should be protected in an unusual degree against the possibility of telescoping in an accident, and the electric apparatus should be installed in such a way as absolutely to prevent the setting of fires. Incombustible metal cars were naturally suggested as the solution of the problem, but in the latter part of 1901, when the question of car design was taken up for the subway, it seemed impracticable to consider the adoption of an all-steel car for the large amount of equipment required, because of the fact that no practical steel passenger cars had ever been constructed and it was evident that to develop a serviceable type a number of very serious mechanical problems had to be attacked by thorough study and experimentation. Not least among the problems was that of keeping the weight of a metal car within reasonable bounds, without sacrificing its strength and serviceability, light weight being an essential requirement in rapid transit operation.

As the best practical solution of the car question, it was, therefore, decided to provide wooden cars initially, but to make them of an advanced type with metal underframes, protected floors and copper-sheathed sides, and to mount the electric apparatus in incombustible envelopes. These cars were rightly considered at the time a great advance upon previous practice in safeguarding against accidental fires. The first lot of 500 of these protected wooden type subway cars was ordered in December 1902.

While it was necessary to insure the operation of the subway at the date set for its opening by providing the initial car equipment, the writer believed the steel car feasible, and in this view he was encouraged by George Westinghouse, under whose stimulating advice he was led to persevere in efforts to develop a metal car at the earliest possible date. A. J. Cassatt, president of the Pennsylvania Railroad, was also impressed with the necessity for non-combustible cars in tunnel service, as the great project of the Pennsylvania road in building a tunnel entrance into New York City was then in progress. He accordingly offered to the subway company the facilities of the Altoona shops to build, in the quickest possible time, a sample steel car, the design of which the writer had completed in October 1902. August Belmont, president of the Rapid Transit Subway Construction Company, concurred in this arrangement and early in 1903 the Altoona shops started upon the construction of this car, which was completed in December of that year. Realizing the many difficulties which would be encountered in getting material promptly at that time, commercial shapes were quite generally used in the design, and the car as built was found, therefore, to be quite heavy and not altogether sightly in appearance.

The company still needed 300 cars to complete the early requirements of the subway operation and it became a question of immediate necessity to determine whether these cars should be of wood or be of the all-steel construction. The writer was able, from experience with the sample car, to develop a new design and at a meeting of the executive committee of the subway construction company early in 1904, he definitely recommended the letting of contract for 200 of the new design of steel cars. On the strong endorsement of E. P. Bryan, general manager of the road, Mr. Belmont decided to venture upon this important innovation in railroad operation. The contract for the 200 steel cars was accordingly let in March 1904, and followed in October of the same year by 100 more. Both these contracts were taken by the American Car & Foundry Company, which had the

courage of its convictions in assuming the heavy responsibility of turning out these large orders at specified time and at a price which was not out of line with that of the previous wooden cars. A number of these cars were received in time for the opening of the subway, October 27, 1904, and are running today.

During the same year the writer, who also had charge of the electrification of the Long Island Railroad, placed an order for 122 steel electric motor cars of practically the same design as the subway cars; this service started June 28, 1905. The Long Island was the first steam railroad in the country to adopt steel cars for its passenger service.

The New York Central a year later placed an order for 125 steel cars and inaugurated their electric service from the Grand Central Station in January 1907.

The Pennsylvania Railroad, as a result of the progressive action of Mr. Cassatt, endorsed by Samuel Rea, then vice-president of the company, adopted steel passenger cars for all trains coming into the new terminal, a decision which has since had a far-reaching effect upon the standards of all railways of the country. The question of the best design for through passenger train cars was taken up exhaustively and systematically by this company and was made the subject of a report by a special committee of its operating officials in May 1909. Today this company has in service 2139 steel passenger cars, excluding a large number of sleeping and parlor cars of the Pullman Company, and builds no other type.

William F. Kiesel, Jr. The method of suspension described by Mr. Summers may be very good on short cars, but with long cars, especially passenger cars, it does not seem sufficiently flexible in the trucks to avoid unbalancing and putting the cars out of shape. The bodies are long and the cars have some flexibility, but in some cases the tracks are such that it is necessary to have excessive provisions for flexibility aside from that in the truck.

John A. Pilcher. Referring to Par. 2 of Mr. Summers' paper, the question of the amount of wind that has to be taken up between the two trucks on the car seems to be exaggerated; in approaching a curve the rise in elevation of the outer rail is about 1 in. in 50 ft. On the ordinary modern passenger car truck centers are about 50 ft. apart so that the total amount of wind is about 1 in. measured at the rails. Considering the car weighs 130,000 lb., with trucks approximately

20,000 lb. each, and the car body about 90,000 lb., in order to take care of the wind in the track, the springs on the diagonals of the car would have to compress ½ in. more than the springs on the opposite diagonal assuming the springs as over the rails. On this same car this would mean that two diagonals would have 20,150 lb., and the opposite two 24,850 lb., or a difference of 4700 lb. This difference in deflection is taken from actual springs.

To analyze this in connection with the swinging hangers, assume these hangers to be 11 in. long, and to be located at an angle with a vertical of 28 deg. 8 min., which is about that shown in the cut, and also assume that they are located approximately over the track (the movement would have to be decreased or increased in proportion to their distance from the rail inside or outside) with a load of 22,500 lb. for each group of hangers.

In order to take care of the same amount of wind in the track as considered in connection with the springs, that is, ½ in. difference in elevation on the opposite sides of the track, the angle would be decreased to 25 deg. 35 min. on one side, and increased to approximately 30 deg. 45 min. on the opposite side in order to bring about stable equilibrium. The vertical loads would amount to 24,935 lb. on one side, and 20,065 lb. on the opposite side, or a difference of 4870 lb., just a little more than when the springs were used. In calculating the deflection of the springs only that of the bolster springs was taken into consideration; the equalizer springs would also have to take an additional load, and would help to reduce the difference of loads necessary to bring about the proper amount of deflection.

Looking at a car from the rear approaching a curve, when the front truck enters the curve the centrifugal force at that point would tend to throw the car body, relative to the truck, actually in the opposite direction from what it should move in order to equalize the stresses. This would put additional torque in the body of the car, which would not be present in the case when springs only take care of this movement. The torque would be rather reduced at the time of entering the curve when the springs only are used.

When both trucks are on the curve all of the wind is out of the car; the centrifugal force in that case throws the car body toward the outside, and would tend to augment the lift in the track on the outside, which is hardly desirable.

Angular hangers, while they may not have been intended for the purpose described, have been in use for a number of years. It is very questionable whether they are of any advantage.

S. A. Bullock. Mr. Pilcher referred to frictionless center plates and adjustable side bearings to reduce to a minimum the oscillation of the car. My experience has been that, to prevent the nosing of a car, which takes place almost entirely upon a tangent, it is necessary to transfer the entire weight from the center plate to the side bearings. Cars of the Pennsylvania Railroad design have been running in the Hudson & Manhattan Subway, and, although they have very short centers, it was found necessary, in order to prevent the nosing of the cars, to take as much weight as possible from the center plates and to put it on the side bearings. All of the weight would have thus been transferred had not the cars been designed with light side sills. It happens in this particular case that the distance center to center of the trucks was exceedingly small, but, even on long steel passenger cars, a saving in wheel flange wear would be effected by taking all of the weight from the center plates and putting it on the side bearings, that is, designing the truck so that immediately the car begins to take or leave the curve, an initial pressure is put upon the truck, which is thus slightly restrained in taking the curve.

This test has been carried out on several railroads. Plaster casts were made of the wheels, and it was found in making outlines of these casts that there is approximately 50 per cent reduction of the wheel flange wear when the radial movement of the truck was restrained in curving.

E. W. Summers.[1] In writing a ten-minute paper, it was not possible to go into detail to any extent. The examples given were intended to be only general in character. Concerning Mr. Pilcher's criticism, the 1-in. wind in track in 50-ft. is the ideal condition. Cars cannot be built to operate only under ideal conditions. Wind in track of 4 in. to 5 in. in the length of a car is frequently encountered when the alignment of the rails is disturbed by water or weather conditions. It is the abnormal conditions that cause wrecks.

In making his comparisons, Mr. Pilcher has apparently neglected the action of springs which are included in the inclined-hanger arrangement. If the inclined hangers make vertical adjustment on account of the tracks being in wind, the springs will not have that to do. As a matter of fact, both the springs and the hangers make some of the adjustment, neither one doing all of it.

As evidence that the vertical reactions given by him are incorrect, compare the ordinary center-bearing truck under a freight car with an

[1]President, Summers Steel Car Company, Pittsburgh, Pa.

inclined-hanger truck such as illustrated in Fig. 1 in the paper, Suspension of Steel Cars, and used under a similar car.

The center-bearing truck must have side bearings, which will be located, say, outside of the wheels in line with the center of the side frame as located on the inclined-hanger truck. Any experienced railroad man knows that side bearings so placed on a center-bearing truck, under such a car, will cause derailment, even on comparatively straight track. On twisted track the weight of the car outside of the wheel uses the wheel as a fulcrum and relieves the load on the opposite wheel, allowing its flange to climb the rail.

It is a matter of record that new refrigerator cars which are comparatively rigid, having side bearings in line with the wheel, easily leave the rails where the track surface is warped.

Contrast these with the inclined-hanger truck having its side bearings outside the wheels and over the center of the side frames under an absolutely rigid all-steel box car, and note that these cars have traversed the worst terminal tracks that could be found at higher speed than the engineman dared to follow with his engine without any indication of wheel lifting, and it is clear that Mr. Pilcher's reactions are in error.

As a comparison with his spring deflection taken from actual springs, some five years ago the writer built an all-steel box car equipped with side bearings directly in line with the truck side frame and M.C.B. springs for a 50-ton car. When attempting to take this car on its own wheels from the riveting shop to the paint shop at the works where it was manufactured it was derailed six times, due to the side bearings being outside of the wheels and warped track surface. The side-bearing, inclined-hanger arrangement was applied to this car, no change whatever being made in the spring arrangement, and the car then traversed the worst tracks to be found around the works and has continued in regular interchange service on the railroads ever since with no indication of derailment or torsional injury to the car body.

# No. 1389

# TEST OF A HYDRAULIC BUFFER

By Carl Schwartz, New York

Member of the Society

The object of this paper is to describe the methods used to determine the performance of an experimental hydraulic buffer for railroad terminal stations and the results obtained; also to illustrate the conditions imposed upon equipment when striking the buffer. It is not intended to enter into the question of design of hydraulic buffers nor to discuss the relative advantages and disadvantages of various means to protect the ends of the railroad tracks against overrunning of trains.

2 The office of a buffer being to bring a locomotive or a train to a standstill when, either by accident or carelessness, it overruns its stopping point, an ideal buffer should be constructed so that during the period of its travel the pressure exerted against the train will be uniform. The buffer will thus absorb the greatest amount of work possible with the smallest maximum resistance against the train, and if it fulfills this condition the reaction will be least harmful to the equipment. In how far the buffer installed in its present form approaches ideal conditions will be shown by the records.

3 The buffer tested consists of a cast-steel cylinder of 22 in. internal diameter, or 380 sq. in. area, and 11 ft. working length. The cylinder is grooved to permit a variable quantity of water to pass by the piston, the amount depending upon the position of the piston, and is largest with the piston drawn out in position to receive a train.

4 The piston proper is attached to a steel ram 10 in. in diameter, extending through a stuffing box, and carrying at its extreme end a head of cast steel with a wooden protection board accurately aligned with the locomotive buffer. The buffer cylinder is connected to city water service, the pressure of which is sufficient to drive the piston out, and the water discharged during the stroke is disposed of to the sewer.

5 The buffer is installed rigidly upon and partly imbedded in a

---

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

block of concrete 20 ft. long, 12 ft. wide and 10½ ft. deep, a total of 90 cubic yards. It is held on each side by five bolts of 2⅜ in. in diameter extending through the foundation into bed rock by a length

FIG. 1 FRONT VIEW OF BUFFER

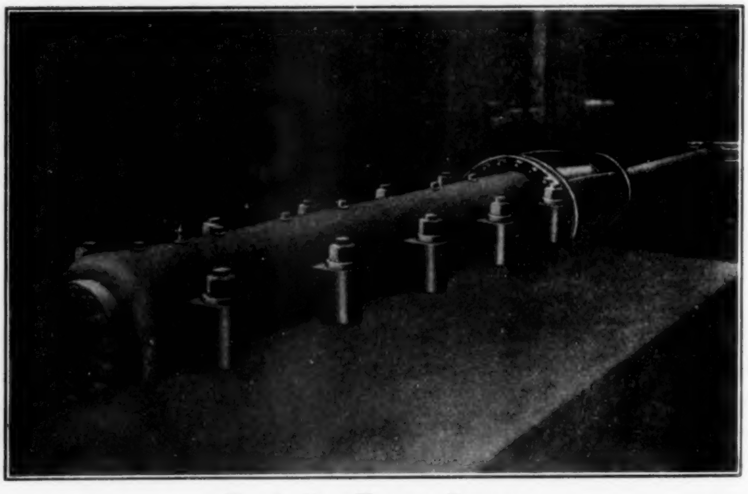


FIG. 2 SIDE VIEW OF BUFFER

varying from 6 ft. in the rear to 13 ft. in the front. The weight of the structure is approximately 390,000 lb. The buffer is illustrated in Figs. 1 and 2.

6 The information required to determine the performance under different working conditions outside of the weight of the train is principally:

*a* Speed of train striking

*b* Pressure performance in cylinder during stroke

*c* Travel of buffer piston.

7 The time of performance being exceedingly short, it was necessary to record the readings automatically, and a special instrument for this purpose was built by an instrument manufacturer on specifications prepared by the writer. (See Fig. 3). This instrument consists of the following parts or mechanisms:

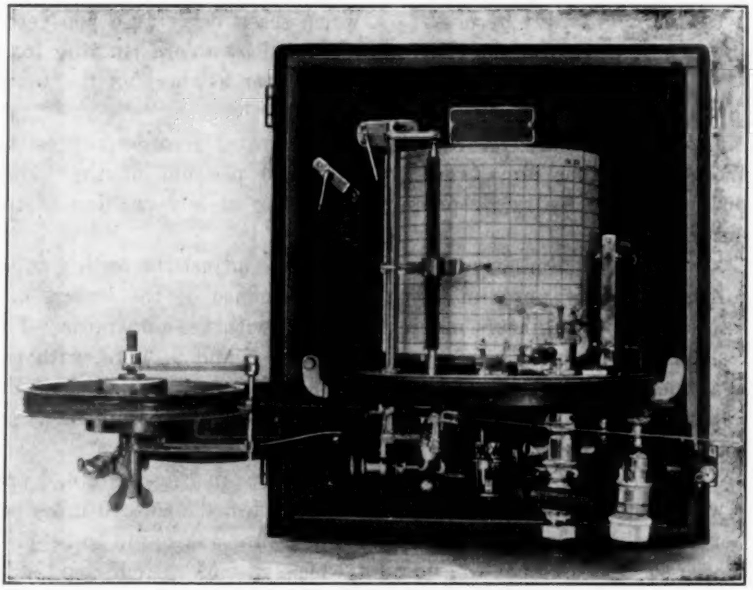


FIG. 3 RECORDING INSTRUMENT

*Recording Cylinder.* A vertical cylinder bearing a recording chart is driven through a worm gear by a small electric motor and can be adjusted by means of a speed regulator to make one revolution in 12 seconds. The circumference of the cylinder being 24 in., 2 in. corresponds to a period of 1 second.

*Train-Speed Recorder.* An electro-magnet moving a pen vertically over the chart in five successive steps. Five contacts

were placed on the track 25 ft. apart in front of the buffer, and these contacts were made and broken by the train and actuated the speed recorder.

*Pressure Recorder.* Constructed like a steam-engine indicator, and by the change of pistons and springs can be used for recording pressures from 0 to 2400 lb. per sq. in. Connection to the rear of the buffer cylinder was made by a small copper tube.

*Piston-Travel Recorder.* A worm screw carrying a pen vertically over the chart and actuated by a cord running from a wheel on a winding spring over a wheel on the worm screw to the head of the buffer.

8 The pressure recorder and piston-travel recorder, operating above each other simultaneously, give the position of the buffer piston and the hydraulic pressure prevailing at any position of the piston.

9 A few preliminary trials were made to adjust the testing apparatus, ascertain approximately the performance of the buffer, and familiarize the engineers making the tests with the apparatus. The tests finally recorded were made on March 8 and 9, 1913, with the equipment and train speeds as given in Table 1. The travel of the buffer piston and the maximum pressure in the buffer cylinder as recorded are also given.

10 From the readings obtained the curves in Figs. 4, 5, 6, 7 and 8 were plotted. In Fig. 4 the highest speed tested was 8.10 miles per hour, at which the maximum cylinder pressure was found to be 1135 lb. per sq. in., corresponding to a total resistance of 431,000 lb., 18,000 lb. of which was balanced by back pressure, leaving 413,000 lb. effective to stop the train. All readings applying to the light locomotive fall almost exactly on the curve and the curve has been extended to show the probable pressure at higher speeds. The readings applying to trains do not coincide as closely with the curve for the reason that the car couplings and the swinging of the cars back and forth had an erratic influence.

11 In Fig. 5 curves *a, b, c* and *d* apply to locomotives and trains of 100 tons, 228 tons, 343 tons and 458 tons respectively and show corresponding maximum piston travels of between 3 and 7 ft. It will be seen that above 5½ and 6 miles per hour the speed of the train has practically no influence upon the travel of the piston; also below 5½ miles per hour the difference in piston travel due to train speed is relatively small.

12 Figs. 4 and 5 illustrate the fact that the impact and pressure against the train depend largely on its speed and that the piston travel is principally a function of the train weight.

13 The curve in Fig. 6 was derived from the preceding and is intended to determine the maximum capacity of the buffer. The highest train weight tested was 458 tons and the extension of the curve shows that a train weight of 1000 tons will drive the buffer piston probably between 10 and 11 ft., or about the total travel for which the buffer is constructed.

TABLE 1 RECORD OF TESTS

| Test No. | Equipment | Weight, Tons | Speed Striking, Miles per Hour | Piston Travel, Ft. | Max. Cylinder Pressure, Lb. per Sq. In. |
|---|---|---|---|---|---|
| 4* | Locomotive | 100 | 4.45 | 2.69 | 460 |
| 5 | Locomotive | 100 | 5.00 | 2.65 | 525 |
| 8 | Locomotive | 100 | 5.30 | 2.70 | 585 |
| 10 | Locomotive | 100 | 6.40 | 2.90 | 730 |
| 1† | Locomotive | 100 | 7.21 | 3.00 | 940 |
| 2 | Locomotive | 100 | 8.10 | 3.00 | 1135 |
| 3 | Locomotive | 100 | 7.12 | 3.00 | 940 |
| 4 | Locomotive | 100 | 7.70 | 3.00 | 1030 |
| 5 | Locomotive, 2 cars | 228 | 4.42 | 4.25 | 490 |
| 6 | Locomotive, 2 cars | 228 | 4.48 | 4.25 | 460 |
| 7 | Locomotive, 2 cars | 228 | 5.37 | 4.30 | 690 |
| 8 | Locomotive, 2 cars | 228 | 6.50 | 4.50 | 790 |
| 9 | Locomotive, 4 cars | 343 | 3.15 | 5.25 | 230 |
| 10 | Locomotive, 4 cars | 343 | 2.90 | 5.20 | 200 |
| 11 | Locomotive, 4 cars | 343 | 4.80 | 5.75 | 515 |
| 12 | Locomotive, 4 cars | 343 | 5.76 | 5.85 | 790 |
| 13 | Locomotive, 6 cars | 458 | 4.50 | 6.56 | 460 |
| 14 | Locomotive, 6 cars | 458 | 5.92 | 6.50 | 820 |

* The first four tests, Nos. 4, 5, 8, 10, were made March 8,1913.
† Tests 1-14 were made March 9, 1913.

14 Fig. 7 covers test No. 1 on March 9 and the curves show the complete performance with a 100-ton electric locomotive running light, as follows:

*a* Speed of the locomotive approaching and during the stroke
*b* Pressure during the stroke
*c* Horsepower absorbed

The area covered by the horsepower curve gives the total energy absorbed by the buffer as 368,000 ft-lb., to which should be added the resistance of the locomotive, calculated at 2400 ft-lb., to obtain a total resistance of 370,400 ft-lb. The energy in the locomotive based upon speed and train weights has been approximately calculated at

370,000 ft-lb., which coincides closely with the resistance recorded. The pressure curve starts with the city water pressure of about 40 lb.,

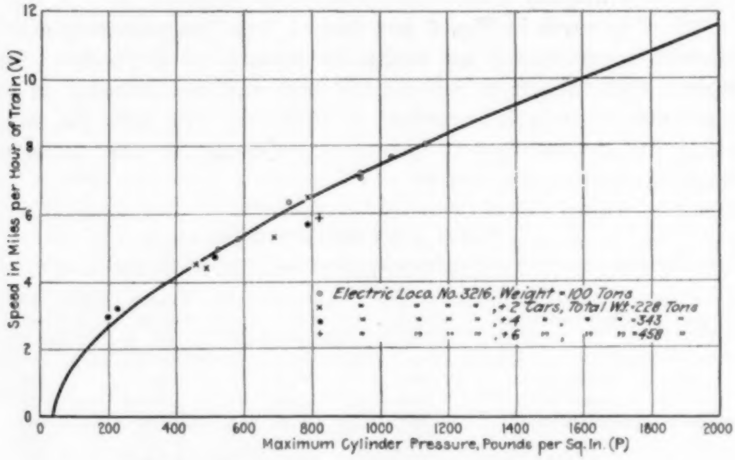



FIG. 4 PRESSURE IN BUFFER AND TRAIN SPEED

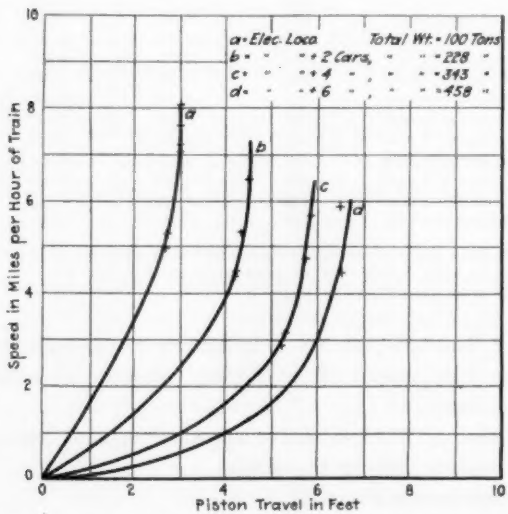



FIG. 5 WEIGHT AND SPEED OF TRAIN AND PISTON TRAVEL OF BUFFER

and was found after the stroke to be about 80 lb., the difference being due to resistance in the discharge valve.

15 Fig. 8 covers test No. 11 and the curves show the complete

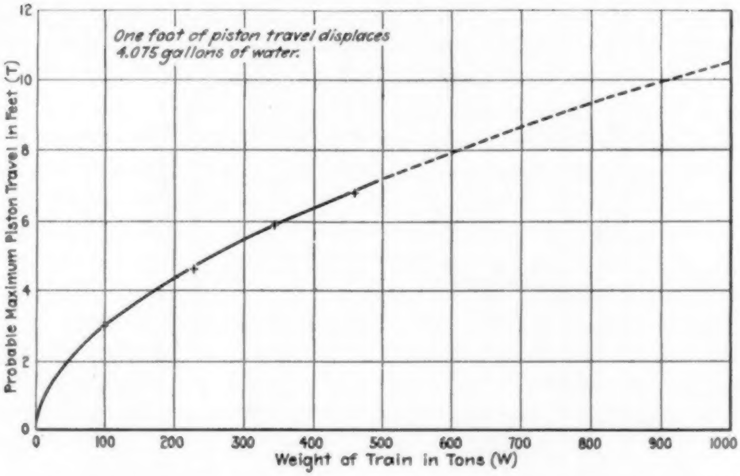



FIG. 6 WEIGHT OF TRAIN AND MAXIMUM PISTON TRAVEL OF BUFFER

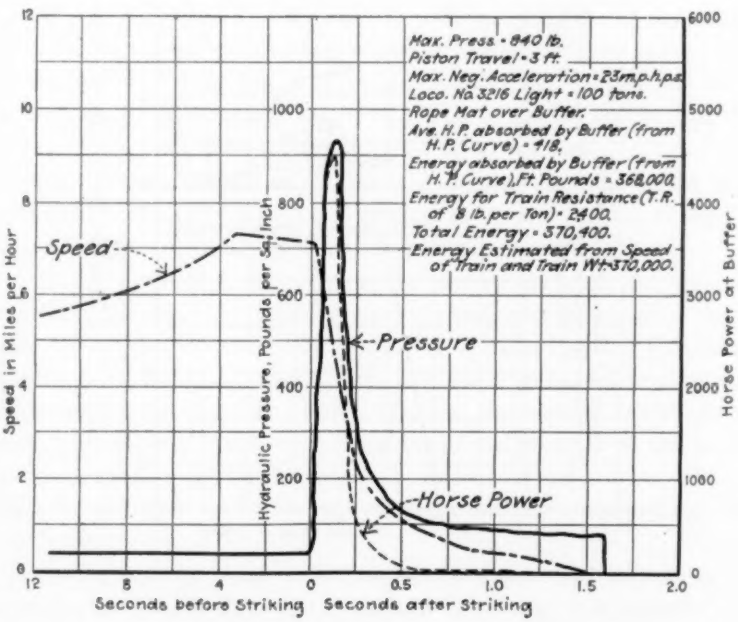



FIG. 7 TOTAL PERFORMANCE—TEST NO. 1, MARCH 9, 1913

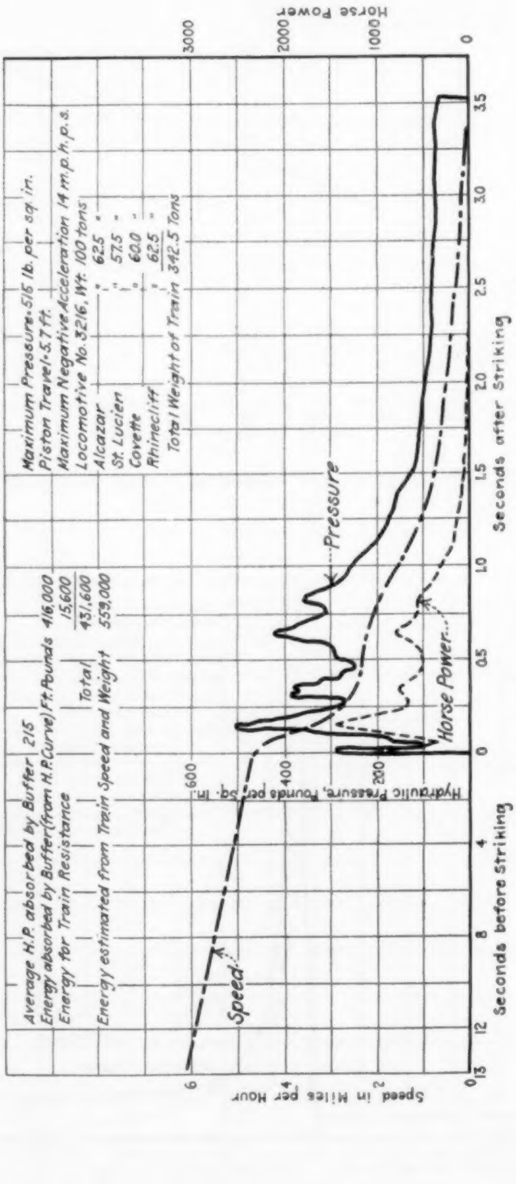



FIG. 8 TOTAL PERFORMANCE—TEST NO. 2, MARCH 9, 1913

performance with a train consisting of a 100-ton locomotive and four pullman cars, the total weight of the train being 343 tons. In comparing this curve with Fig. 7, the following should be noted :

16 The maximum pressure is only about 500 lb. instead of 900 lb. because the speed of the train was only 4.8 miles per hour instead of 7.2 miles per hour. The peaks in the pressure curve are probably due to the locomotive and cars striking separately about as follows: Locomotive buffer 300 lb., locomotive body 500 lb., first car 390 lb., second car 300 lb., third car 420 lb. and fourth car 360 lb.; but evidently the train was drawn together and pulled apart, which makes the performance somewhat irregular. Comparison between the energy absorbed by the buffer and the energy in the moving train shows a discrepancy of about 128,000 ft-lb., or roughly, 25 per cent, which can be accounted for as energy absorbed in the train by its parts swinging back and forth during the impact.

17 Other tests were calculated as the results given in Figs. 7 and 8 and show similar and consistent performance.

## CONCLUSIONS

18 Referring to Fig. 7, it will be seen that the bulk of the energy is absorbed during the first ¼ second of the stroke; the impact was considerable after striking, the pressure falling off immediately after exceeding the maximum. Fig. 8 would show similar results had the speed of the train been higher than 4.8 miles per hour.

19 It was demonstrated during the tests that the buffer was sufficiently effective to prevent damage to the locomotive or equipment though the speeds were at times relatively high.

20 It is evident that the impact can be made smaller by distributing the pressure uniformly over the period of the stroke. To do this the leakage in the buffer should be increased at the beginning of the stroke to reduce the initial peak in the pressure curve at speeds exceeding, say 4 miles per hour. This will increase the travel of the piston for a given train weight and reduce the capacity of the buffer to some extent. If the leakage is brought into definite relation to the pressure curve the buffer should offer a uniform resistance against the train.

21 In how far these conditions can be approached in practice is a matter of investigation and the writer hopes that this contribution may be of assistance in showing the conditions to be fulfilled.

## DISCUSSION

F. H. Clark. One of the important elements in any device intended to absorb the shock or arrest the speed of moving bodies is the length of stroke or the distance through which the device operates. The disadvantage of the ordinary type of buffer or stop as used in railway service is the relatively short movement of the face of the buffer. In the hydraulic buffer described by Mr. Schwartz, this difficulty is overcome to a great extent by the stroke or working length of 11 ft., and the tests naturally show that desirable results are obtained. The device has the disadvantage, however, of occupying a considerable amount of space. The length over-all is not given, but it would probably not be less than 30 ft., or about 20 ft. more than that occupied by the usual type of buffer. This would be an important item in a large terminal, not only on account of the additional space required, but also on account of the greater distance between the train and the station.

Buffers as a rule are intended only for emergency use and are very seldom brought into action. It is generally desirable that equipment of this sort be of such construction that casual inspection may determine whether or not it is in working order, and in cold climates, there might be some liability of the device freezing or becoming otherwise inoperative without its condition being noticeable.

Arthur E. Johnson. This buffer could be so designed as to be similar to the simpler forms of brakes used to take up the energy of the recoil of guns, as follows:

First by ascertaining as nearly as possible the amount of pressure which a given train construction will stand as a resistance. Then the length of action of the buffer can be arranged as permitted by the space available and as required by the weight of trains. Cylinders may be arranged either horizontally or vertically to save space. They may be provided with a by-pass if necessary to handle trains very much lighter than the maximum. The lighter resistance thus provided would permit the same length of action for the light as for the heavy trains. And last, solid bumpers must, as before, be provided to prevent the pistons from bottoming whenever the energy is too great to be taken up entirely in the device.

Oil would seem to be a much more suitable fluid for this use than water on account of the greater viscosity.

H. A. Jensenius.[1] In traveling through Germany and England last summer, I saw a buffer in Düsseldorf and also in England, the construction of which was similar to the one described by Mr. Schwartz, but the stroke, I believe, was not more than 8 ft. At crowded terminals space is very precious, and therefore if an apparatus of this kind could be built with a little shorter stroke and probably with a larger cylinder or higher pressure, it might be better adapted for the purpose intended.

Another point is the question of freezing. These buffers are filled with liquid and subject to low temperatures. Therefore, water would not be suitable. A device arranged with a closed pressure tank so as to confine the liquid and not to require any outside source of supply would, in my judgment, be more serviceable. In this case oil or any other non-freezable fluid could be used.

Philander Betts. The curves showing the results of tests of the stoppage of trains made by the use of this apparatus show the great importance of the draft gear. I would like to know, if the question can be answered, whether note was made as to the types of draft gears and whether tests have been made with different types of draft gear to show the relative results in the stoppage of trains as indicated by the form of curve drawn by this apparatus.

The Author. The length of travel of the buffer piston is largely influenced by the weight of the train striking the buffer and relatively little by the speed of the train, the speed of the train determining the pressure in the cylinder and thus the resistance against the train. The buffer is built for a maximum train weight of 1000 tons, and with about 11 ft. travel will be able to bring such train to a standstill. The length of the stroke could be shortened by increasing the pressure against the train with an increased liability of damage to the equipment.

Mr. Jensenius is correct in stating that buffers installed in Germany and England have a shorter working length, and the reason is that trains in Europe have smaller tonnage than in this country. The experimental buffer tested is the largest of this type ever constructed, its total length being 30½ ft.

The suggestion is made to use a liquid other than water. Inasmuch as the energy absorbed is all transformed into heat, from a theoretical standpoint water would be the ideal medium because it is the medium

[1]R. D. Wood & Co., Philadelphia, Pa.

capable of storing the largest number of B.t.u. Practically, however, this feature is of little importance and the non-freezing qualities of oil or glycerine are more valuable. Water was used and disposed of to the sewer because the buffer is installed for experimental purposes, but for a permanent installation the suggestion to use a closed pressure tank to confine and circulate the liquid should receive careful consideration.

The tests were made irrespective of different types of draft gears and the curves in Fig. 8, which apply to a locomotive with four Pullman cars and a total train weight of 342.5 tons, show that a certain amount of energy was absorbed by the train itself due to the cars swinging back and forth during the impact. No doubt the type of draft gear has some influence upon the behavior of the train during the impact period.

No. 1390

# PATENT LICENSE RESTRICTIONS

## THE PATENT LAW AS AFFECTED BY RECENT AND PENDING DECISIONS OF THE SUPREME COURT OF THE UNITED STATES AND PROPOSED LEGISLATION

By Edwin J. Prindle, New York City

Member of the Society

The Supreme Court of the United States on March 11, 1912, rendered a decision in the case of Henry vs. Dick which has been generally misunderstood and has aroused clamor for changes in the patent law of great public importance. The court sustained the right of a patentee to dictate conditions under which his patented machine could be used, and the same reasoning would maintain his right to fix prices below which his product could not be sold. To many laymen and some lawyers the decision seems contrary to the Sherman anti-trust act.

2 The Supreme Court has also since rendered a decision in what is known as the Bath Tub case, which bears on the relation of the patent law to the Sherman act, and tends to mark out the line between them.

3 A case known as Bauer vs. O'Donnell is now before the Supreme Court for decision on the right of the patentee to dictate the price at which his article may be sold by the retailer.

4 There is a general misunderstanding of the issues involved in Henry vs. Dick, and most men, upon hearing an adequate explanation of what the decision means, see that the rights of the public are not in any manner jeopardized by it. In consequence of this situation, and particularly of the decision in the case of Henry vs. Dick, many bills have been introduced in Congress seeking to modify the patent law in fundamental particulars, and not only to cut down the monopoly of the inventor but to put his property at a great disadvantage compared with unpatented property, and the

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

situation is a serious one. It is the object of the present paper to explain the meaning and effect of the decisions in question and of the proposed legislation.

### HENRY VS. DICK

5 The case of Henry vs. Dick will be discussed first and will be treated with what might otherwise appear to be disproportionate fullness because it will be used as a basis for explaining most of the principles involved in the controversy.

6 Dick invented a mimeograph or copying machine and patented it. He sold it with this label prominently displayed on the machine:

LICENSE RESTRICTION

**This machine is sold by the A. B. Dick Company with the license restriction that it may be used only with the stencil paper, ink and other supplies made by the A. B. Dick Company, Chicago, U. S. A.**

7 A Miss Skou bought one of the Dick machines and Sidney Henry, the defendant in this case, sold her a can of ink, intending that the ink should be used on the Dick machine, and knowing that such use would be in violation of the notice on the machine. The Supreme Court held that Miss Skou had no right to use the machine with ink bought from any one else than the Dick company, and that Henry, in aiding her intentionally to violate her obligations to the Dick company, was a contributory infringer and was properly enjoined from a repetition of his act.

### REASONS FOR ENACTING THE PATENT LAW

8 Before the patent law was enacted, the only way in which an inventor could obtain a return from his invention was by keeping it secret; otherwise a competitor might copy the invention and without any overhead expense or initial investment of capital, as compared with the inventor, reap the benefit of perhaps a large expenditure of time and money made in producing and perfecting the invention. As a result, comparatively few inventions came into existence, and those that did were jealously kept secret so far as possible. Workmen were sworn to secrecy, and the inventions were practiced only under lock and key and with every effort to prevent their leaking out.

9 Under these circumstances, the public could not get the full benefit of the invention, because the inventor could not manufac-

ture in as large quantities nor as cheaply as if it had been unnecessary to observe these precautions. If there was an ingredient or a formula or some particular step in a process which one person could take care of in the manufacturing, the inventor was likely to keep the knowledge of that one point entirely to himself, intending, usually, to disclose it to some one else when death approached, but when, as often happens, death came without warning the inventor died without disclosing the secret, and the invention was lost to the world forever unless some one else rediscovered it. There are not a few of these lost secrets or "lost arts."

10 When the patent law was enacted it said, in effect, to the inventor: "If you will give us (the public) a complete disclosure of your invention so that anyone skilled in the art can practice it, we will give you an absolute monopoly of this invention for seventeen years. This will enable you to practice the invention without fear that it will be stolen, so that you can get the full harvest from it during the life of your patent, and we will have the benefit of it forever afterwards." The fathers of our country regarded this law as so important that they made provision for it in the Constitution itself. The first patents were granted by a board consisting of George Washington, his secretary of state and his attorney general, after a personal hearing of the inventor, and they personally signed these patents.

### WHAT A PATENT GRANTS

11 The patent law simply says to the inventor: "We will give you a clear field to make money out of your invention if you can. Whether you ever make a dollar or not depends upon your own ability and industry in making, using and selling your patented invention." The patent does not cost the public one cent, for the inventor even pays all the expenses of the Government in granting the patent.

12 The patent, therefore, grants to the inventor the right to exclude or enjoin all others from any making, using or selling of his patented invention. He may either get his return by keeping the market wholly to himself and manufacturing and selling the invention, or he may sell to others immunity from the injunctions to which the inventor is entitled by his patent. In other words, he may sell licenses or other forms of grant under his patent, which means, in effect, that "For the consideration which you pay me, I give up my right to exclude you from my monopoly to the extent we agree upon."

13 The patentee is under no obligation either to manufacture the patented invention himself or to let anyone else manufacture it. The Supreme Court has said, by a unanimous decision in the Paper Bag case, that it is the patentee's right to sit still during the life of his patent and neither manufacture himself nor let anyone else manufacture. This exclusive monopoly is the reasonable price which the public agreed to pay for the making and disclosure of the invention, and during the life of the patent it must let the invention alone. When the term of the patent has expired, the public will have an unrestricted right to use the invention of which, but for the inventor, it would never have known.

### DICK'S RIGHTS UNDER HIS PATENT

14 Dick, then, was at liberty to prevent anyone from using his patented machine during the life of his patent. That would have been to exercise his *entire* right of exclusion. He did not choose to do this, however. He might have sold his machine outright for a sum of money representing the entire profit to which he thought he was entitled for his invention. If he had charged more than the public thought the machine was worth, he would have made no sales. The public would have been under no obligation to purchase the machine. Dick found that his machine did not work successfully with every kind of ink, nor with every kind of stencil paper, but that it worked much better with a certain kind of ink, and with stencil paper having certain qualities. It was to Dick's interest that the machine should do good work, in order that he might sell a large number of machines, and it was also to the interest of the public that the machines should not be disappointing to the users. Dick considered that if he only *partially* released each machine from his power of injunction, to the extent of requiring that the ink and stencil paper be bought of him, he would be able to make certain that the users of it had the right sort of ink and paper. He would also be able to sell the machine for a comparatively low price at the outset and add his profit a little at a time to the ink and stencil paper. In this way the initial cost would not be burdensome, and the price that a person paid for a machine would be in proportion to the amount of use which he had of it; that is, in proportion to the amount of ink and stencil paper which he used upon it.

15 Another case in which the Supreme Court upheld the right of the patentee to say that his patented article should only be used under certain conditions is the Cotton Tie case, in which case the

court decided that the patentee of a buckle for fastening bales of cotton could lawfully impose the condition on the purchaser that the buckle should be used only once. The patentee stamped on each buckle the words "licensed to use once only." In removing the buckle from the bale it was necessary to cut the buckle. A rival manufacturer bought the buckles after they had thus been used and cut, and repaired them and offered them for sale again. He, like Henry, in the Dick case, was restrained.

16 As Dick, under the Paper Bag decision, had the power to say to the public, "You shall not use my machine at all during the life of my patent," it was reasonable that he should have the power to say, "You may use it, but only on certain conditions," providing the purchaser agreed to those conditions at the time he purchased the machine.

17 *A sale of a patented article without conditions puts it forever beyond the control of the patentee.* It has been repeatedly decided that if an inventor once sells a machine without any conditions attached to the sale, that machine passes beyond, or without, the monopoly, and he can never again exercise any control over it whatever. It is a misunderstanding on this point that has caused much of the cry against the decision in this case. If Miss Skou had bought the machine without any license restriction on it, or without such restriction being called to her notice in any way, she could have used it with any ink she chose without interference from Dick. When she bought the machine with the license restriction displayed on it in such a way that she could not fail to notice it, she impliedly consented to the conditions of that license restriction and bought the machine subject to them. Having bought the machine with this *qualified* title to it, she could not object to Dick's wanting the bargain lived up to. The physical machine belonged to her; that is, the metal, the screws, and other parts, and she was at liberty to make any use of them which did not infringe the monopoly granted by the patent, but she was not at liberty to trespass upon the portion of the monopoly which Dick reserved to himself when he sold her the machine; namely, the right to use that machine with other inks and other stencil papers than those made by Dick.

18 Many people think that this decision gives the patentee the right to pursue the purchaser of his patented machine and interfere with its use or sale after he has received his full pay for it, but that is not so. If he once sold the machine free from his monopoly, it would be gone forever so far as he was concerned. It is only when

he sells it with the distinct understanding, either expressly agreed to by the purchaser or impliedly agreed to, because the understanding is stated in a notice fixed on the patented article so prominently that it cannot be reasonably overlooked, that he can exercise any control whatever over the patented article in the user's hands. It is not conceivable that an injunction would be granted by any court in a case where the printed notice on the patented article was not so prominently displayed as to be certain to attract the purchaser's attention.

19 Ex-President Taft, when on the bench, participated in a decision which upheld rights similar to those in the Dick machine. This related to machines for fastening buttons on shoes, the machines being sold under the license restriction that the purchaser should use them only with button fasteners bought of the manufacturers, as follows:

CONDITIONS OF SALE

**This machine is sold and purchased to use only with fasteners made by the Peninsula Novelty Company to whom the title to said machine reverts upon violation of this contract of sale.**

20 These button fasteners were not patented, and the manufacturers took their profit on the machine chiefly through charging a larger price for them than the market price. The payment for the machine was entirely inadequate, considered by itself, to compensate even for the machine. The court said that the button fasteners were, in effect, counters by which the payment for the machines were measured. This case was not appealed to the Supreme Court, but that court referred to it in a later case as a "leading case," which is a strong expression of legal approval.

21 *Henry could not have been enjoined unless he knew that his ink was to be used on the Dick machine.* The injunction was issued against Henry in this case only because he intentionally helped to bring about a violation of the rights reserved by Dick and consented to by Miss Skou under Dick's implied contract with Miss Skou. It has been repeatedly held that one who intentionally helps another to infringe a patent can be enjoined as a contributory infringer. This is precisely what Henry did. He sold the ink to Miss Skou, knowing that she was under an implied contract not to use the ink on the machine, and Henry intended that she should use it on the machine. If Henry had merely sold Miss Skou the ink, having no reason to suppose that she was going to use it in violation

of a license, he would not have been an infringer and could not have been enjoined. There is absolutely no foundation for any fear that this decision would render any dealer liable to injunction who sells ink, or any other supply, without any reason to suppose it is going to be used in violation of a patent.

22 *The decision does not give Dick a monopoly of ink.* There has been fear on the part of many people that the effect of decisions like the Dick decision and the button fastener machine decision would be to give the patentee a monopoly of unpatented products; for instance, in the one case of inks, and in the other case of unpatented button fasteners. This is erroneous, however, as the public is just as free to buy ink for any other purpose than for use on the Dick machine as it ever was, and so, also, with the button fasteners. It is only when ink is bought or sold from others than Dick for the specific purpose of use with the Dick machine that Dick can interfere under his patent. It is impossible that this control over the inks to be used with Dick's machine should affect the price of ink for other purposes. Dick could not control the price of one-tenth of one per cent of all the ink sold in this country, for all of his machines together would not use that amount of ink. Whatever market for ink Dick controlled was of his own creation, by inventing the machine which uses the ink.

## THE BAUER VS. O'DONNELL CASE

23 This case is closely connected in principle with the case of Henry vs. Dick. The Bauer Chemical Company manufactures a food tonic, and sells this tonic with the following label pasted on each package:

> For sale and use at a retail price not less than one dollar ($1). Any sale in violation of this condition, or use when so sold, will constitute an infringement of our patent, under which [the tonic] is manufactured, and all persons so selling or using package or contents will be liable to injunction and damages.
>
> A purchase is an acceptance of this condition. All rights revert to the undersigned in the event of violation.

24 A Washington druggist bought the preparation from the manufacturers and sold it at a price below $1. He also bought it through jobbers and cut the price. Upon his failure to abide by promises that he would not cut the prices, the company owning the patent brought suit against him under the patent for an injunction to restrain him from cutting the price and for an accounting of profits

and damages. The Court of Appeals for the District of Columbia certified this case to the Supreme Court of the United States for instructions, and the case is now awaiting a hearing by the latter court. This is the first case which has ever brought squarely to the Supreme Court the question of the patentee's right to dictate the price at which his product shall be sold, although the question is involved in a number of other suits now pending in the lower courts, and has been many times decided by the lower courts in this country and in England. The decision of the Supreme Court will, of course, settle the law on this point in the United States.

25 As explained in connection with Henry vs. Dick, the patentee really has three separate monopolies: The right to exclude all others from making, from using, and from selling his invention. These rights, with one very recent exception, have been treated by the lower courts in the United States as separate and distinct and capable of being separately sold. They have also each of them been held to be capable of sale for a limited territory, or for limited times, or limited to special trades or arts.

26 In Henry vs. Dick, the Supreme Court sustained the right of the patentee to prescribe the conditions under which his patented product should be used. In the Bauer vs. O'Donnell case, the question before the Supreme Court will be whether the patentee can prescribe conditions under which his article can be sold. The question is very important to patentees because the whole advantage of the patent would be lost in some instances if it were not for this power over the sale of the article.

### THE DICK DECISION NOT IN CONFLICT WITH THE SHERMAN ACT

27 While this right of the patentee to maintain the prices of his patented article, and Dick's control over the ink, may appear a violation of the Sherman act, in reality it is not. The Supreme Court has expressly decided that the control of the patentee over his patented article is not a violation of the Sherman act. The reasons will be apparent upon considering the purposes of the two acts. The patent act was for the purpose of inducing inventors to make inventions and to put the public in possession of a working knowledge of them, and the sole inducement was a monopoly granted to the patentee for a limited time to make money out of the invention, if he could.

28 On the other hand, the Sherman act was for the preservation of the rights of the public in trade which it already possessed and

to prevent raising by a monopoly the prices of articles in which the public already had a right to trade.

29 The patent statute was for the purpose of bringing into existence trade in articles which never had and perhaps never would have existed but for the monopoly offered by the public as a reward to the inventor for inventing them, while the Sherman act was to preserve to the public free competition in trade which already existed and belonged to the public. The patentee takes no rights from the public when he restricts the conditions of use or price of sale of patented inventions. He is but dealing with his own. He can suppress it if he wishes. The public can have it without price simply by waiting until the patent has expired, but the patentee has a contract (his patent) giving him the exclusive right until that time. The public never had any right to sell ink for use with Dick's machine, for the machine never existed until he invented it. Therefore he has taken nothing away from the public which belongs to it, and there is nothing for the Sherman act to operate upon.

30 The control of the patentee over the conditions of use or sale of his patented article is not so absolute as it might seem. The only way in which he can get return from his patented article is by inducing the public to use it. If, then, he imposes restrictions which are too burdensome or which take away the advantage of the public in using the article, he cuts down the sales which he might otherwise make. This fact alone puts a limit on the restrictions which the patentee will impose on his patented article.

### THE BATH TUB DECISION

31 The patent involved in the bath tub decision was one relating to an implement for shaking sand, more or less automatically, on a red hot iron bath tub for the purpose of forming an enamel on the bath tub. Under the guise of licensing manufacturers under the patent to use the implement, a *combination of manufacturers* was affected, which prescribed prices and conditions under which bath tubs were to be sold. The *association* forbade the selling of "seconds" or bath tubs which were in any manner imperfect, and not only fixed the prices, but imposed penalties for selling below those prices. The *association* also provided a jobber's license agreement which he had to execute before he could purchase the licensed sanitary enamel ware. Various zones were established which were to be preserved to specified parties. There were other regulations too numerous to mention.

32 The Supreme Court in the Bath Tub case held, in effect, that the manufacturers had formed a combination in restraint of trade and in violation of the Sherman act, and that the patent was a mere cloak. The vital objection was that the manufacturers formed a combination *between themselves.* The court did not decide that the patentee could not have lawfully imposed the same conditions separately upon each manufacturer as a license if there was no combination and conspiracy between the manufacturers. If the manufacturers had all had their relations directly with the patentee and there had been no combination between themselves, so far as the Supreme Court decided, it would not have been objectionable. The court did not decide whether or not it was lawful to regulate the price of unpatented bath tubs under a patent for a patented implement used in the manufacture of the bath tubs.

### THE PROPOSED PATENT LAW REVISION

33 Following the Dick decision a flood of bills was introduced into Congress seeking to amend the patent statute to cut down the inventor's monopoly and greatly restrict his control over the property which he had created. The theory of the framers of these bills was that this property was going to come into existence anyhow, no matter how little the inducement, and it was perfectly safe to regulate it to any extent, any monopoly which remained in the inventor being mistakenly thought to be a free gift from the public instead of an inducement, without which the invention could not come into existence.

34 The principal bill, which was recommended by a majority (only) of the Patent Committee of the House of Representatives, contains almost every negative amendment of the patent law that anybody has suggested. This bill has not yet passed either house of Congress. The three principal provisions of the bill are:

*a* Compulsory licenses which are to be obtainable from a Federal District Court by anyone who can establish that the owner of the patent (not being the original inventor) has purchased the patent for the purpose of suppression. The Patent Committee of the House of Representatives was unable to point out any tangible instance of such patent suppression, and only an average of one to six out of about sixty witnesses who testified before the Committee, or who presented communications on the subject, were in favor of any such provision. A

complete discussion of the provision would be beyond the scope of this paper, but a few of the main objections may be stated as follows:

If this amendment were made, no manufacturer would dare to patent or disclose anything but the preferred form of the invention, because inferior forms which he might patent and disclose could be manufactured by others, under the compulsory license, in competition with his preferred form. This might destroy the effect of a generic or parent patent which was parent to the children. It is unlikely that a court would require the licensee to pay a price commensurate with the expense to the manufacturer of the cost of developing the preferred form, so that the manufacturer would be at a disadvantage. The manufacturer could not afford to improve his product, because the moment he began to manufacture the improved form, somebody would demand the right to manufacture the poorer, original form. No manufacturer could afford to own more than one patent in a given line. The constant danger of litigation to compel a license would make patents much less desirable property than they are today, and greatly decrease their value. Thus the effect of the compulsory license would be to discourage invention instead of to encourage it. It is believed that there are very few instances of suppressing the best invention in a particular line, and so long as a manufacturer furnishes the public with the best form of the invention, it is to the benefit of the public to give him a monopoly during the years which it will take to establish the article thoroughly in the trade.

*b* The bill also forbids license restrictions as to the manufacture, use or sale of his patented product by taking away practically all power to enforce any such restrictions, even though the purchaser agreed to them in order to induce the purchase. This again would very greatly lessen the value of patents and consequently reduce the incentive to make inventions, so that inventions would not be made to the same extent as today. The bill forbids the patentee the control over his property which largely resides in the owner of unpatented property. It puts the inventor

at a disadvantage as compared with the owners of other property.

*c* The bill also provides that a presumption that the Sherman anti-trust act has been violated shall be conclusively presumed, when any one of a large number of the most common business transactions is entered into in connection with patented property, regardless of any surrounding circumstances. If the patentee attempts to restrict the price at which the article may be resold; if he attempts to compel a purchaser to buy an unpatented article from him with the patented article (such as in the Dick case); if he purchases other patents with a view to preventing competition with his patented article; if he attempts to control the territory in which his article shall be sold; or to sell to one person on conditions less favorable than to another; or to do business under any other name than his own, or that of his firm or corporation, he is conclusively presumed to have violated the Sherman anti-trust act, and anyone who has been injured may bring a suit within three years after he has been damaged and recover three-fold damages, the costs of the suit, and his attorney's fees. Moreover, the patent may be declared forfeited and the owner of the patent may be fined $5000 and imprisoned for a year. In other words, the patentee is tremendously worse off with his valuable property, which he has created with his own brain, and which is a great benefit to the public, than the owner of any other kind of property.

35 *There is grave danger in putting the knife to the inventor's reward.* The temptation may be strong to prohibit the patentee from exercising such control over the sale or use of his patented article as the Dick decision gives. It looks so easy simply to amend the patent statute and cut down the extent of the monopoly granted by the patent. It seems as though the public might just as well have this advantage as not. But, as a practical matter, it is to be considered whether the public would not lose more than it would gain. Taken as a whole, the price is a mere bagatelle which the public has paid for the inventions that have been produced solely as a result of the patent statute. The patents have not cost the public a single penny, for the inventors have, by their fees, paid all the cost to the Government of granting them. On the other hand, the price paid

by inventors in the cost of experiments, and in the time involved and the energy expended, has been beyond all calculation.

36 There is one machine in the shoemaking art upon which over $500,000 was expended before it was considered perfect enough to put on the market. Instances are not at all rare where $100,000 or more has been expended in developing an invention. Most of the large progressive manufacturing concerns of the United States maintain a corps of inventors who are seeking to improve their products and cheapen the cost of manufacture, all of which goes into the general fund of human knowledge, and, after the short period of the patent has passed, is forever at the service of anyone who wishes to use it. Not only have the patented inventions cost the public absolutely nothing, but the inventor will get no return whatever, unless he can succeed in making money out of the invention. This means that he must *further* serve the public by supplying it with the patented product in order that he may have any return.

37 *The patent statute has been a cardinal factor in the development of our country.* No one statute in all history has made for the material advancement of mankind to an extent at all comparable with the patent statute. Man's hands have been multiplied and made more efficient and skilful by the inventions produced as a result of the patent statute, so that the average man today lives with a degree of comfort, a power of communication and transportation, a quantity of reading matter, a variety of pleasures and a control over nature which were unobtainable by the man of 100 years ago, regardless of his wealth.

38 Let us take the tools and appliances which the inventor has supplied the farmer as a typical instance. In 1791, the best plow was a wooden structure shod with iron, and it was so imperfect that but an acre of land could be plowed in a day, and even then the soil was not much more than scratched. The plow had hardly been improved at all in 40 centuries. Now, a steel plow turns up the ground so much deeper and better that a much larger crop is grown, and several times as much work can be done in a day with the ordinary one-horse plow as with the old form, while with the steam-driven gang plow 30 times as much work can be done in a day. In George Washington's day thrashing was done by the use of a flail, producing about five bushels of wheat a day. Now, a steam thresher can thresh a thousand bushels in the same time, and there are even steam-propelled harvesting machines which cut a swath 26 ft. wide through a field of wheat, and thresh, clean and sack the grain at the rate

of three bushels a minute, and yet but seven men are required to run such a machine. All these changes have come about as the result of the American patent law, and practically within the space of less than a century. If there were time to review the similar changes in other fields of industry due to our patent law, it would read like a fairy tale. Many arts and industries owe their very creation and existence to the patent law.

39 *The inventor's reward should not be disturbed, but, instead, the patent law should be simplified and strengthened.* With the bargain already so tremendously one-sided in favor of the public, and with the burden already so heavy on the inventor, we should be exceedingly slow to lessen the inducement to the inventor to continue his work. At the very longest, his monopoly can exist but 17 years. A considerable part of this period is usually taken up in getting the invention successfully upon the market, even in the case of the most successful inventions. The necessity he is under to sell his product is a natural limitation on the conditions which he may impose. At the most, the public need only wait the 17 years in order to have perfect freedom to make, use and sell the invention.

40 It would be unwise, in the extreme, to jeopardize the effective continuance of this most advantageous work on the part of the inventor by cutting down in any way the return to him which, in any case, costs the public nothing, and is a trifle compared with the benefit to mankind. The effect of any present change in the patent laws would not become evident for a considerable length of time, and it might take many years to recover the lost ground, not only in the degree to which we have fallen behind invention abroad, but in the loss of inventive power due to incentive to develop it.

41 The patent law needs changes, but changes to build it up, not to tear it down. The Patent Office should have a sufficient force of adequately paid examiners so that it can make practically certain it is issuing patents only for inventions which are new. The procedure of adjudicating a patent should be made cheaper, simpler and shorter. For that purpose there should be enough judges so that patent cases may be quickly tried by testimony taken orally in the presence of the judge, instead of by printed records of testimony taken slowly and expensively before a master and then read before a judge. The fees now payable to the clerks of the courts should be reduced to merely a fair compensation for the services performed. And a single court of patent appeals should be established, instead

of the present intolerable system of nine separate circuit courts of appeals, each having equal and independent jurisdiction in their own circuits of every patent.

## DISCUSSION

J. Nota McGill.[1] Mr. Prindle's paper should awaken the members of the Society to a realization of the necessity for vigorous protest against the enactment of dangerous legislation aimed at the fundamentals of our patent system, and which, if enacted, will be a fatal step backward and tend seriously to impair that set of laws which have done more for mankind physically, socially and financially, than any laws on our statute books.

The growth of that system has been co-extensive with the growth of our country; it can truthfully be asserted that to the growth of the inventive genius of our people is due America's great progress. The increase in population in practically the last 60 years of the 19th century was about 530 per cent; the increase in patents about 6400 per cent. At the beginning of the 19th century the world knew nothing of telegraphy, telephony, steam navigation, or the myriads of labor-saving devices, the introduction of which has revolutionized the social, financial and commercial world. At that time agricultural products constituted our export trade. Manufactured products were nearly all imported, as it had been England's policy to suppress all industries of this kind in the colonies.

The American inventor could, with justification, claim much at the hands of his country; he asks only for fair treatment. In the grant of patents for invention there is no undue advantage bestowed upon any man; the inventor is given the right, for a limited time, to *exclude* others from practising his invention, on the condition and with the express understanding that he shall disclose and make known to the world what he has produced, so that at the expiration of that limited period the public may be free to use his invention without payment or tribute.

Until a few short months ago no serious complaint was heard against our patent system. Then suddenly came the proposed amendments. If we look for a cause it is difficult to find, unless it be the methods adopted by the Shoe Machinery Trust. But even granting that their methods were the cause, it is a safe assumption that had there been no dissenting opinion in the Dick case we would not today

[1] Secretary, Patent, Trademark and Copyright Section, American Bar Association, Washington, D. C.

be confronted with menacing legislation. There has been no public clamor for a change. The pronouncement in the Dick case involved no new proposition of law. The decision in the Button Fastener case was rendered in 1896, and for 16 years thereafter (nearly the lifetime of a patent) the commercial world accepted the doctrine of that case practically without question, and the public suffered no hardships. The public are not required to take what the patentee offers if his terms are unreasonable. If they subscribe to the conditions imposed it is but a tribute to the value of the invention. If, on the other hand, they are not accepted, the public are deprived of nothing to which they had any legal or moral right. The patentee is alone the sufferer if in consequence of his attitude he is without a market, and this fact alone precludes him from ever imposing unreasonable demands. The American public can be depended upon neither to require nor to accept unfair conditions in affairs of business.

The other radical change now advocated (compulsory licenses) is even more dangerous and less justifiable than the curtailment of the right to dictate the terms of use and sale.

It has been said that the patent privilege "encourages the inventor to bring his invention to the highest possible condition of practical utility by inventing improvements on it constantly, in order to keep pace with the public wants and to control the trade from which his compensation is derived." But this stimulus to improve will no longer exist if the patentee is to be penalized because of the improvement.

Mr. Edison stated to the committee of the House that while he had heard and read numerous statements that many corporations buy valuable inventions to suppress them, he did not know of a single specific case of suppression. When Mr. Fish asked the committee if during the hearings or at any time there had been brought to the attention of the committee a single specific case of suppression, he was told that *one man* had stated instances where his own inventions had been suppressed. Considering the fact that over a million patents have been granted in this country, the complaint of a single or even a hundred or more persons is practically negligible.

It is unreasonable to assume that the American manufacturer would for a moment refrain from placing on the market an invention that would insure a fair return on the investment. American capital has never been known to indulge in such pastime as is suggested by a handful of persons demanding privileges which the law has always reserved to the inventor and those claiming under him.

Once suffer such an amendment to be made, and a serious blow will be struck the inventors of our country. Capital, ever apprehensive, will hesitate to accept the risk entailed; the inventor will find it more difficult to secure support; and progress will be arrested if the cost of adopting an improvement may mean an advantage to competitors, royalty or no royalty.

Action and untiring action alone will avert the impending dangers pointed out by Mr. Prindle. Concede to the members of the committee of the House the best motives, they are not actuated by any desire other than to discharge their duties honestly and faithfully. It is within the power of the members of this Society interested in maintaining the integrity of our patent system to impress upon every member of Congress that the proposed changes are not desirable, but on the contrary will prove a decided menace.

The author did not desire to present a closure.—Editor.

## ADDENDUM BY THE AUTHOR

As the foregoing paper was written in March 1913, the following is added to indicate briefly the changes which have taken place in the decisions and the Legislative situation between that date and February 1, 1914:

The Supreme Court has since decided the case of Bauer vs. O'Donnell, holding that a patentee may not prevent a retailer from cutting the price on a patented article by attaching thereto a label stating that the article is licensed for sale at retail under a patent at a price not less than a specified price. This decision was rendered by a majority of but one of the nine Justices of the Court, so that it is possible that future decisions of the Court upon analogous but somewhat different states of facts, may be in favor of the patentees. The Court mentioned various conditions as absent from the state of facts in Bauer vs. O'Donnell, thereby seeming to imply that if they had been present, the decision would have been favorable to the patentee. The decision is, therefore, not to be regarded as absolutely denying the patentee all right under all circumstances to control the retail price at which the article shall be sold, but only under the conditions then before the Court. The decision is said to have affected goods whose annual sales amounted to approximately one billion and a half dollars a year.

The United States Circuit Court of Appeals in New York (doubtless following the decision of the Supreme Court in Bauer vs. O'Donnell) affirmed the decision of the lower Court in the suit of Waltham vs. Keene, denying the right of the Waltham Watch Company to compel Keene, a retailer, to sell their movements only at specified retail prices. There have also (since Bauer vs. O'Donnell) been decisions of United States District Courts between other parties to the same effect.

The Legislative Bill, described in Section 34 of the paper, has been reintroduced into the House of Representatives at the present session of Congress in substantially the same form as described in the paper.

A bill known as the Kahn Act slipped through Congress practically without public notice, and without public hearings, whose object it is to make the Panama-Pacific Exposition at San Francisco more attractive to foreign exhibitors by giving such exhibitors a monopolistic protection to their articles which are exhibited at the Exposition for three years after the close of the Exposition. The bill became a law on September 18, 1913, making it unlawful to "copy, imitate, reproduce or republish" anything exhibited at the Exposition which is protected by a foreign patent, copyright, trademark, etc., without the authority of the "proprietor" thereof. A branch copyright and patent office is to be maintained at the Exposition and required to give the "proprietor," free of charge, a certificate amounting to legal evidence of such "proprietorship." The rights are to be enforced by injunction, the assessing of damages and profits, the surrender of alleged infringing articles to be held during the suit, the surrender of articles found to infringe, and all means of making them to be destroyed. The law makes lawful infringement a penal offense, punishable by imprisonment for a year or less, or a fine of $100 to $1000, or both. The effect of the Act is to grant a patent, trademark, copyright, etc., to every exhibitor who holds a foreign patent, trademark, copyright, etc., from the time his goods are exhibited to December 4, 1918, and longer if the Exposition should be prolonged, and the penalties are far more severe than any penalties for infringing any copyright, patent or trademark obtained of the United States Government in the first instance.

This law is exceedingly dangerous and unjust. The United States Government will grant patents only for new inventions, while many of the foreign governments grant patents without any examination whatever; that is, they grant patents upon mere application, even though the invention be as old as the hills. The Kahn Act requires the United States Government to give the same force to such foreign patents, as it does to its own patents, which are only granted for new inventions. Thus, a United States manufacturer who could not obtain a patent in the United States for his product, can send to France, and obtain a French patent without the slightest difficulty, and then by exhibiting his article at the San Francisco Fair, the American manufacturer can obtain a certificate of "proprietorship" from the Branch Patent Office at the Fair, and not only shut out his competitors from what they have a perfect right, under our patent law, to make, but put them in prison for exercising that right.

The scope of this footnote will not permit further discussion of this Act, but enough has been said to indicate its effect. There is no need for such a law, for foreigners to have the same rights as United States citizens to obtain United States patents for inventions and designs, and to obtain copyright and trademark protection. The Act was never even presented to the Patent Committee of either House of Congress, and there is a strong movement on foot to secure its repeal or modification.

No. 1391

# COST OF UPKEEP OF HORSE-DRAWN VEHICLES AGAINST ELECTRIC VEHICLES

By W. R. Metz, Washington, D. C.

Member of the Society

During the fall of 1910 the writer and the accountant of our office were instructed to submit a report as to the desirability of purchasing motor trucks to replace horse-drawn wagons, and an investigation was made covering certain Government departments and private firms. This report was submitted in November 1910, and it was estimated that the office would save approximately $11,000 per annum if electric vehicles were purchased and all of the horse-drawn equipment sold.

2 This report was approved and equipment purchased as outlined herein, and the results were most gratifying as the saving during the first year was nearly $12,000, in spite of the fact that during this year, only a part of the electric truck equipment was purchased and six horses were kept. Five of these horses have since been sold, and the saving during the next year will undoubtedly be increased.

## PRELIMINARY INVESTIGATION

3 The naval gun factory at Washington operated five electric trucks, two of which had been in operation for four years; one of these had a capacity of 2500 lb., and the other a capacity of 5 tons. The first was furnished by the McCrea Motor Company of Ohio, and the other by the Studebaker Company. The cost of operation and the saving accomplished were as given in Table 1.

## FIVE-TON ELECTRIC TRUCK

4 The cost of operating this truck was about the same as the 2500-lb. wagon, excepting that the 2500-lb. wagon cost 75 cents for charging per 40 mile radius while the 5-ton truck cost $1.10 for the same radius. The total cost of operation is given as $2396.84, and

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

if depreciation and interest on the investment of $3725 are included the total cost will be $2843.84.

TABLE 1 2500-LB. ELECTRIC TRUCK

| | | | |
|---|---|---|---|
| Cost of Truck | $2230.00 | | |
| Labor for charging batteries | | $46.44 | |
| Charging | | 16.50 | |
| Acid | | 18.00 | |
| Rubber jars | | 15.00 | |
| Batteries (partly renewed) | | 64.98 | |
| Carbon brushes | | 1.80 | |
| Repairs | | 99.96 | |
| 1 Operator at $2.48 per day | | 776.24 | |
| 2 Laborers at $1.92 per day each | | 1201.92 | |
| Totals | $2230.00 | $2240.84 | |
| Depreciation 10 per cent | | 223.00 | |
| Interest on investment at 2 per cent | | 44.60 | |
| Total cost | | | $2508.44 |
| Total Mileage per Year | 3366 | | |
| Cost per mile | $0.745 | | |
| This truck displaced 5 horses and carts, costing as follows: | | | |
| 5 Carts by contract at $1.92 per day | | $3004.80 | |
| 5 Laborers at $1.92 per day each | | 3004.80 | |
| Total | | $6009.60 | |
| Net Saving of Truck over Horses per Year = | $3501.16 | | |

5 This truck displaced two 2-horse wagons costing as follows:

| | |
|---|---|
| 3 Laborers, each wagon, at $1.92 per day | $3605.76 |
| Stabling, shoeing, etc., 4 horse at $4 per day | 1252.00 |
| Total | $4857.76 |
| Net Saving of Truck over Horses | $2460.92 |

In this case depreciation and interest are not included as the original cost of the horses was not known.

6 One large company operates a number of horse-drawn wagons, also a number of motor trucks, both electric and gasolene driven. Their experience is given in Table 2.

7 From Table 2 can be estimated the yearly expense of vehicles requiring more than one horse, by adding the items for horses and harness to the one-horse vehicle costs as follows:

| | |
|---|---|
| 1-horse vehicle expense | $464.81 |
| Vehicle and 2 horses expense $464.81 plus $275.71 | 740.52 |
| Vehicle and 3 horses expense $740.52 plus $275.71 | 1016.23 |
| Vehicle and 4 horses expense $1016.23 plus $275.71 | 1291.94 |

These costs do not include labor and stable expense.

TABLE 2 HORSE-DRAWN VEHICLES

| | | |
|---|---|---|
| **Investment** | | |
| 1 Horse | $250.00 | |
| 1 Vehicle | 125.00 | |
| Harness | 30.00 | |
| Total | $405.00 | |
| **Maintenance and Upkeep** | | |
| Depreciation | | |
| Horse at 20 per cent | $50.00 | |
| Vehicle at 15 per cent | 18.75 | |
| Harness at 15 per cent | 4.50 | |
| Interest on $405 at 6 per cent | 24.30 | |
| Total | | $97.55 |
| **Horse Upkeep** | | |
| Feed at 47.4 cents×365 days | $173.01 | |
| Shoeing at 7.5 cents×365 days | 27.38 | |
| Veterinary at 1.1 cents×365 days | 4.02 | |
| Total | | 204.41 |
| Vehicle Expense at 43.4 cents per day | $158.35 | |
| Harness Expense | 4.50 | |
| Total | | 162.85 |
| Total Expense exclusive of Labor and Stable | | $464.81 |

8 In Table 3 are given the costs of operation for the electric trucks used by this same company. Table 4 compares the operation costs of the first two trucks recorded in Table 3 on the basis of 300 days' service per year. Table 5 makes a similar comparison of the commercial and individual gasolene machines assuming 300 days' service per year for the former and 365 days for the latter. The following makes of gasolene cars were in use: Cadillac, Brush, Buick, Ford, Franklin and Maxwell, ranging from 10 h.p. to 30 h.p.

9 In addition to the records obtained from this company, the attempt was made to secure data from a number of private firms, but unfortunately none of them had any exact figures and could only approximate the savings due to the use of both electric and gasolene-driven trucks over horse-drawn vehicles.

## VEHICLES IN USE BY OFFICE

10 During the fiscal year 1910 and 1911, which were the last two years that horse-drawn vehicles were used exclusively, the total expenses for the stable were $31,113.58 for 1910, and $31,231.93 for

TABLE 3 EXPENSE OF OPERATING ELECTRIC COMMERCIAL VEHICLES

| Capacity, Lb. | 850–1000 | 1500–2000 | 2500–3000 | 4000 | 7000 |
|---|---|---|---|---|---|
| Interest and Depreciation (Machine less Batteries and Tires)............ | $244.50 | $306.30 | $391.40 | $422.94 | $470.84 |
| Mechanical and Electrical Upkeep.............. | 67.54 | 84.15 | 101.70 | 110.96 | 121.42 |
| Tire Repairs and Renewals.................... | 79.28 | 97.30 | 155.05 | 267.60 | 535.25 |
| Battery Repairs, Cleaning and Renewals......... | 130.50 | 175.36 | 219.34 | 271.54 | 312.84 |
| Current at 1 cent per kw-hr.................... | 20.00 | 30.20 | 40.00 | 60.00 | 51.50 |
| Totals............ | $541.82 | $693.31 | $907.49 | $1133.04 | $1491.85 |

TABLE 4 TOTAL MILEAGE OF THE FIRST TWO CARS, AND THE COST PER MILE OF EACH, EXCLUSIVE OF INTEREST AND DEPRECIATION

| Capacity, Lb. | Miles per Day | Miles per Year | Cost per Year | Cost per Mile |
|---|---|---|---|---|
| 850 to 1000 | 25 | 7500 | $297.32 | $0.04 |
| 1500 to 2000 | 25 | 9125 | $277.01 | $0.03 |

TABLE 5 COST AND MILEAGE OF COMMERCIAL AND INDIVIDUAL MACHINES

| Service | Miles per Day | Miles per Year | Total Cost per Year | Cost per Mile |
|---|---|---|---|---|
| Commercial | 22.2 | 6660 | $896.69 | $0.135 |
| Individual | 19.7 | 7190 | 809.53 | 0.1125 |

TABLE 6 COMPARATIVE EXPENSE OF OPERATING MOTOR VEHICLES IN PLACE OF HORSE-DRAWN VEHICLES

| | | |
|---|---|---|
| Vehicles for Emergency Work | | |
| Two 2-horse vehicles at $740.52 | $1481.04 | |
| One 3-horse vehicle at $1016.23 | 1016.23 | |
| | | $2497.27 |
| One 4000-lb. automobile | $1133.04 | |
| One 2500-lb. automobile | 907.49 | |
| | | $2040.53 |
| Annual Saving by Use of Automobiles, not including Labor | | $456.74 |
| Supply Wagon | | |
| One 4-mule wagon | $1291.94 | |
| One 4000-lb. automobile | 1133.04 | |
| Annual Saving | | $158.90 |
| Paymaster's Money Wagon | | |
| One 2-horse vehicle | $740.52 | |
| One 850 to 1000-lb. vehicle | 541.82 | |
| Annual Saving | | $198.70 |

1911; and for the same years the cost of the delivery section was $17,093.93 for 1910, and $17,256.19 for 1911, making a total cost for the delivery and stable sections of $48,207.51 for 1910, and $48,488.12 for 1911. It should be stated here that the stable and delivery sections were entirely separate at this time, each being in charge of a foreman, whereas after the automobiles were used these sections were combined, although the costs were separated.

11 The equipment, number of men employed, and the expenses of the stable and delivery sections during the fiscal year 1910 are given in Table 7.

12 Omitting the wages paid the drivers and messengers, the stable section alone cost $18,447.37, making the cost per horse per year $802.06, or $2.20 per day, of which 37¾ cents was for feed.

13 From Table 7 can readily be obtained the operating cost of one 2-horse, 5000-lb. capacity wagon, and these are given in Table 8.

14 During the month of November 1911, there were purchased and put into service two 1000-lb. trucks, two 2000-lb. trucks, and two 5000-lb. trucks, all of the electric type, and at the same time 17 horses and their equipment were sold, leaving six horses, one 5-ton capacity, two-horse truck, one single-horse truck, one carriage driven by two horses, and one carriage driven by one horse.

15 In November 1912, one additional 5000-lb. truck and an

TABLE 7 EQUIPMENT, NUMBER AND CLASS OF MEN EMPLOYED, AND COST OF OPERATION, MAINTENANCE AND REPAIR FOR HORSE-DRAWN VEHICLES FOR YEAR ENDING JUNE 30, 1910

| Equipment | Cost |
|---|---|
| 23 Horses (average per year) | $6,900.00 |
| Harness, blankets, etc | 1,350.00 |
| 1 Five-ton truck (2 horse) | 425.00 |
| 7 Large delivery wagons (2 horse at $475.00 average) | 3,325.00 |
| 6 Single delivery wagons (1 horse at $275.00 average) | 1,650.00 |
| 3 Light mail wagons at $200 | 600.00 |
| 4 Depot wagons (carriages) at $300 | 1,200.00 |
| 2 Coupes, with pole and shafts, at $540 | 1,080.00 |
| Total | $16,530.00 |

Number and Class of Men Employed

| STABLE SECTION | DELIVERY SECTION |
|---|---|
| 1 Foreman at $2000 per year | 1 Foreman at $2000 per year |
| 1 Mail carrier at $3.20 per day | 4 Helpers at $3.04, 2.80, 2.72 and 2.40 per day |
| 1 Messenger at $2.40 per day | 12 Messengers at $2.40 per day |
| 11 Drivers at $2.40 per day | 2 Unskilled laborers at $2.00 per day |
| 1 Stableman in charge at $2.88 per day | |
| 8 Stablemen at $2.40 per day | |

Total employees[1] in both sections = 42.

| Cost of Operation of Stable Section | |
|---|---|
| Wages of foreman and stablemen | $10,113.59 |
| Wages of drivers | 12,666.21 |
| Rent | 2,400.00 |
| Feed | 3,172.20 |
| Supplies | 959.62 |
| Repairs to harness, wagons, etc | 626.86 |
| Shoeing | 906.60 |
| Gas and electricity | 268.50 |
| Total Cost of Operation, Maintenance and Repair | $31,113.58 |
| Depreciation, Horses 20%, Harness 15%, Wagons 10% | 2,410.50 |
| Interest on Investment at 2% | 330.60 |
| Total Cost including Depreciation and Interest on Investment | $33,854.68 |

| Cost of Operation of Delivery Section | |
|---|---|
| Salaries and wages | $17,085.93 |
| Material and supplies | 8.00 |
| Total Cost | $17,093.93 |

[1] 20 per cent additional is allowed for night service.

electric-driven carriage were installed, and two carriage horses and one truck horse and their equipment were sold.

16 In January 1913, one 8000-lb. capacity truck was installed, and two more truck horses and their equipment were sold.

17 The truck equipment, and the number and class of men employed for caring for the electric vehicles and the remaining horses during the year 1912 is given in Table 9.

TABLE 8 COST PER YEAR OF ONE 2-HORSE, 5000-LB. CAPACITY WAGON

| | | | |
|---|---|---|---|
| Wagon Expense | | | |
| Cost of truck | $425.00 | | |
| Cost of maintenance and repair | | $31.75 | |
| Cleaning and washing | | 66.76 | |
| Lubricants | | 1.00 | |
| Depreciation, estimated at 10% | | 42.50 | |
| Total | | | $142.01 |
| Horse Expense | | | |
| Cost of two horses | $615.00 | | |
| Cost of feed | | $264.33 | |
| Cost of care (hostler) | | 702.60 | |
| Cost of veterinary and office labor | | 250.10 | |
| Cost of medicine | | 2.17 | |
| Cost of shoeing | | 78.84 | |
| Cost of blankets, nets, etc | 12.22 | | |
| Rental value of space (2 horses) (based on $2400 for 22 horses) | | 218.00 | |
| Depreciation, estimated at 20% | | 125.44 | |
| Total | | | $1641.48 |
| Harness Expense | | | |
| Cost of harness | $123.00 | | |
| Cost of maintenance | | $7.68 | |
| Depreciation, estimated at 15% | | 18.45 | |
| Total | | | $26.13 |
| Miscellaneous Supplies | | $10.00 | |
| Drivers' Wages | | 751.20 | |
| Helpers' Wages | | 751.20 | |
| Gas and Electricity | | 23.34 | |
| Interest on Investment at 2% | | 23.50 | |
| Total | | | $1559.24 |
| Total Original Cost | $1175.22 | | |
| Total Expense for One Year | | | $3368.86 |

18 During the coming year, and after the remaining six horses are sold, the four stablemen will be dispensed with and in their place

there will be employed probably two helpers and one laborer at a total cost of $6.80 per day, or $2128.40, thus indicating a further reduction of $1377.20.

19 All of the electric trucks were purchased from the Baker Motor Vehicle Company, of Cleveland, Ohio, and the iron-clad exide storage batteries from the Electric Storage Battery Company, of

TABLE 9 EQUIPMENT AND NUMBER AND CLASS OF MEN NOW EMPLOYED

| | | |
|---|---|---|
| Equipment | | |
| Two 1000-lb. trucks | $4,639.00 | |
| Two 2000-lb. trucks | 5,498.78 | |
| Three 5000-lb. trucks (two in use during full year) | 10,625.22 | |
| One 8000-lb. truck (installed in January 1913) | 5,509.00 | |
| One electric-driven carriage (installed in November) | 3,671.00 | |
| Total Cost | | $29,943.00 |
| Number and Class of Men Employed | | |
| 1 Foreman at $2000 per year | $2,000.00 | |
| 12 Chauffeurs at $2.40 per day each | 9,024.40 | |
| 3 Helpers at $3.20, $2.80 and $2.72 per day | 2,729.36 | |
| 10 Messengers at $2.80 (two) and $2.40 (eight) per day | 7,762.40 | |
| 1 Mail carrier at $3.20 per day | 1,001.60 | |
| 2 Messenger boys at $1.20 each per day | 751.20 | |
| 1 Stableman in charge at $4.00 per day | 1,252.00 | |
| 3 Stablemen at $2.40 each per day | 2,253.60 | |
| 2 Drivers at $2.40 each per day | 1,502.40 | |
| 35 | | $28,276.96 |

Note—Labor charge is based on day service.

Philadelphia, Pa.

20 It will be noted that there have been no tire costs during the year. This, of course, was due to the fact that none of the tires have worn out, but it should be stated that the tires on the 5000-lb. trucks are practically worn out and, in fact, were being replaced during the month of January 1913, so that the tires on these trucks wore out after running since December 1911, or a period of 13 months. The tires on the 1000 and 2000-lb. trucks are still in fairly good condition. The tires on the 5000-lb. trucks would undoubtedly have lasted longer than they did were it not for the fact that these trucks were often loaded up to 7000 lb., making a considerably harder service than could have been anticipated by the makers.

21 The total cost of operating the combined delivery, stable, and garage during the year 1912 was as given in Table 10.

22 In Table 10 it is interesting to note that the six electric trucks, doing practically all of the work, cost only about twice as much as the six horses. As a matter of fact the six electric trucks alone did as much work as the total stable force did during the year previous, and the horse-drawn equipment simply took care of the increased work.

TABLE 10 SALARIES, WAGES, LEAVE OF ABSENCE, HOLIDAY AND LIABILITY PAYMENTS, MATERIAL AND SUPPLIES, REPAIRS, ETC., FOR DELIVERY, STABLE AND GARAGE, JANUARY 1, 1912 TO DECEMBER 31, 1912

| Month | Salaries, Wages, Leave of Absence, Holiday and Liability Payments | | | Material and Supplies, Repairs, Etc. | | |
|---|---|---|---|---|---|---|
| | Delivery | Stables | Garage | Delivery | Stables | Garage |
| January | $1,429.23 | $523.71 | $818.23 | $14.75 | $149.85 | $223.87 |
| February | 1,330.59 | 465.88 | 822.56 | 2.74 | 127.06 | 224.89 |
| March | 1,345.76 | 487.61 | 828.97 | 2.28 | 189.36 | 240.67 |
| April | 1,245.10 | 484.17 | 891.61 | 13.51 | 76.21 | 197.08 |
| May | 1,378.52 | 543.23 | 1,019.64 | 25.55 | 114.27 | 183.87 |
| June | 1,141.73 | 453.61 | 821.78 | 66.96 | 213.18 | 495.99 |
| July | 1,454.96 | 512.77 | 887.06 | 9.40 | 124.35 | 173.57 |
| August | 1,372.38 | 447.99 | 889.27 | 6.00 | 127.84 | 251.64 |
| September | 1,487.09 | 429.50 | 678.68 | 4.34 | 122.55 | 198.60 |
| October | 1,391.65 | 443.33 | 710.59 | 9.94 | 129.54 | 307.26 |
| November | 1,565.02 | 337.95 | 809.68 | 1.35 | 57.76 | 222.40 |
| December | 1,460.70 | 252.00 | 910.13 | 13.95 | 48.81 | 203.41 |
| Total | $16,602.73 | $5381.75 | $10,088.20 | $170.77 | $1480.78 | $2923.25 |

Equipment issued January 1 to December 31, 1912 $70.00 $24,446.00

23 Table 11 shows in detail the cost of taking care of the electric trucks, and as the cost of the trucks of any given size is about the same, only one of each size is given.

24 From the given figures we can readily ascertain the cost of operating one 5000-lb. capacity electric truck as compared with one 5000-lb. capacity 2-horse wagon. The cost of two 2-horse wagons being $6737.72 (Table 7), the saving due to the use of the electric truck is $4204.39.

25 Unfortunately no mileage record of the horse-drawn vehicles was kept during the previous years, but the records show that the two-horse wagon made an average of four trips per day, and the

TABLE 11 ELECTRIC VEHICLE RECORD

| | 1000-Lb. | | | 2000-Lb. | | | 5000-Lb. | | |
|---|---|---|---|---|---|---|---|---|---|
| | Jan.-June | July-Dec. | Total | Jan.-June | July-Dec. | Total | Jan.-June | July-Dec. | Total |
| Total Miles | 3013.00 | 3511.00 | 6524.00 | 2978.00 | 2864.00 | 5842.00 | 3586.00 | 2918.00 | 6504.00 |
| Ampere Hours | 8598.00 | 8459.00 | 17057.00 | 11019.00 | 9710.00 | 20929.00 | 18302.00 | 14096.00 | 32398.00 |
| Ampere Hours per Mile | 2.85 | 2.40 | 2.61 | 3.69 | 3.39 | 3.58 | 5.10 | 4.83 | 4.98 |
| Kw-Hr. Supplied | 1249.8 | 1303.80 | 2553.6 | 1573.6 | 1389.4 | 2963.00 | 2678.4 | 2047.8 | 4726.2 |
| Cost of Current | $14.67 | $16.64 | $31.31 | $18.58 | $17.79 | $36.37 | $31.56 | $26.28 | $57.84 |
| Battery Upkeep | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| Tire Upkeep | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... | ...... |
| Electrical Upkeep | 1.20 | 2.93 | 4.13 | 0.75 | 14.38 | 15.13 | 5.06 | 8.67 | 13.73 |
| Mechanical Upkeep | 26.80 | 27.68 | 54.48 | 41.34 | 28.41 | 69.75 | 129.62 | 80.10 | 209.72 |
| Inspection | 161.58 | 138.67 | 300.25 | 161.58 | 138.67 | 300.25 | 161.58 | 138.66 | 300.24 |
| Totals | $204.25 | $185.92 | $390.17 | $222.25 | $199.25 | $421.50 | $327.82 | $253.71 | $581.53 |

present records show that the 5000-lb. electric trucks make an average of eight trips per day. The mileage of the electric trucks being about 24 per day, it may be assumed that the mileage of the horse-drawn trucks was about 12 per day.

TABLE 12 COST OF OPERATING ONE 5000LB. ELECTRIC TRUCK[1]

| | | |
|---|---|---|
| Cost of Truck and Equipment (including spare battery and parts) | $3745.00 | |
| Maintenance and repair (as per statement above) | | $581.53 |
| Depreciation at 10% | | 374.50 |
| Interest on investment at 2% | | 74.90 |
| Chauffeur's wages at $2.40 per day | | 751.20 |
| Messenger's wages at $2.40 per day | | 751.20 |
| Total original cost | $3745.00 | |
| Total expense for one year | | $2533.33 |

[1] One of these electric trucks easily replaces two 2-horse trucks.

26 For the sake of comparison the costs of the three sizes of electric trucks and of the two-horse wagon are given in Table 13.

TABLE 13 COSTS OF ELECTRIC TRUCKS AND 2-HORSE WAGON

| | 2-Horse Wagon 5000 Lb. Capacity | 5000-Lb. Electric Truck | 2000-Lb. Electric Truck | 1000-Lb. Electric Truck |
|---|---|---|---|---|
| Average Trips per Day | 4 | 8 | 8 | 9 |
| Mileage per Day, Average | 12 | 24 | 20 | 20 |
| Mileage per Month (Loaded Halfway) | 312 | 624 | 520 | 520 |
| Average Load per Trip, Lb. | 4000 | 5500 | 2500 | 900 |
| Total Load per Day, Lb. | 16,000 | 44,000 | 20,000 | 8100 |
| Tons | 8 | 22 | 10 | 4.05 |
| Total Load per Month, Tons of 2000 Lb. | 208 | 572 | 260 | 105.3 |
| Total Cost per Month | $280.74 | $211.11 | $187.81 | $180.93 |
| Cost per Mile | 0.899 | 0.338 | 0.361 | 0.347 |
| Cost per Mile (Omitting Driver's and Helper's Wages) | 0.499 | 0.138 | 0.121 | 0.107 |

[1] One of these electric trucks easily replaces two 2-horse trucks.

The average load on the 5000-lb. electric truck is 5500 lb., that on the 2000-lb. truck is 2500 lb., that on the 1000-lb. truck is 900 lb., and that on the 2-horse wagon was 4000 lb.

27 The methods of keeping the records are given as a matter of interest. The Daily Meter Record and the Daily Charging Record are made out, one for each truck, by the employees in the garage. These are turned in to the office each morning and transferred to the

Electrical Vehicle Record each month. This card is printed on both sides, thus providing space for a whole year.

28 The electrical and mechanical upkeep are obtained from the regular mechanics' work slips. Every machine in the office is given

GOVERNMENT PRINTING OFFICE
ELECTRICAL SECTION

DAILY METER RECORD—ELECTRICAL VEHICLE

*Machine No*...... *Battery No*...... *Date*................................, 191

| | READING OF ODOMETER | READING OF AMP. HR. METER | TIME | REMARKS |
|---|---|---|---|---|
| IN........... | ............ | ............ | ......... | ............ |
| OUT......... | ............ | ............ | ......... | ............ |
| DIFFERENCE.. | ............ | ............ | ......... | ............ |

FORM S—450

(*Signed*)..........................................................

GOVERNMENT PRINTING OFFICE
ELECTRICAL SECTION

DAILY CHARGING RECORD—ELECTRIC VEHICLES

*Battery No*.......... *Date*.............................. 191

| TIME | VOLTS | AMP. | SP. GR. | TEMP. |
|---|---|---|---|---|
| ......... | ............ | ............ | ......... | ......... |
| ......... | ............ | ............ | ......... | ......... |
| ......... | ............ | ............ | ......... | ......... |

FORM S—447

(*Signed*)..................................................

a number, and all repairs made and materials used are entered on the work slip and charged against the machine number and tabulated in the office. In this way an accurate record is kept as to the cost of each machine.

29 Under inspection is charged the time for charging batteries, washing the trucks and similar miscellaneous work in the garage, and all of this cost is evenly distributed among all of the trucks in service.

FORM S-406

GOVERNMENT PRINTING OFFICE

OFFICE SUPT. OF BUILDINGS

ELECTRICAL VEHICLE RECORD MACHINE No. ............

| MONTH | JAN. | FEB. | MAR. | APR. | MAY | JUNE |
|---|---|---|---|---|---|---|
| TOTAL MILES | | | | | | |
| AMP. HRS. USED | | | | | | |
| AMP. HRS. PER MILE | | | | | | |
| K. W. HRS. SUPPLIED | | | | | | |
| COST OF CURRENT | | | | | | |
| BATTERY UPKEEP | | | | | | |
| TIRE UPKEEP | | | | | | |
| ELECTRICAL UPKEEP | | | | | | |
| MECHANICAL UPKEEP | | | | | | |
| INSPECTION | | | | | | |
| TOTALS | | | | | | |

REMARKS ..........................................................................

..................................................................................

..................................................................................

..................................................................................

## DISCUSSION

H. H. SMITH.[1] Scientific costkeeping is unfortunately a very modern practice, and we who must displace outworn institutions with new and better ones are at a disadvantage because there is nothing tangible available concerning the old.

The saving effected by the substitution of power trucks in the case cited by Mr. Metz is surprising, as it is not unusual for the cost of operation during the first year to be reduced very little. Usually this is because the delivery routes and system have not been altered where necessary to fit the new conditions. Obviously if an electric truck is given the work of a horse truck on the horse-truck schedule, the cost per delivery will be much higher because the advantages of the motor-driven vehicle will not have been properly utilized. The case under

[1]Edison Storage Battery Co., Orange, N. J.

discussion indicates what may be accomplished by careful study and rearrangement.

Another point brought out in the paper which confirms other investigations along the same lines is the insignificance of the cost of energy in the operation of electric vehicles. It seems to run generally between 6 and 1 per cent of the total cost. Repairs and depreciation usually constitute a greater item of expense, while the cost of labor ranges from 40 or 50 per cent up. These relations should be carefully borne in mind when the purchase of trucks is under consideration.

It is an interesting fact, brought out more clearly in other investigations, that the advantage of the electric wagon over the horse wagon increases with the distance of the delivery zone from the starting point in light parcel delivery, and with the weight of the load in work near the point of distribution. It is even recorded that the horse may be of slightly less cost in congested territory where the radius of action is small and the number of stops large.

It is stated in the abstract of the paper that possibly the gasolene truck meets with favor in certain classes of service because of the ease with which fuel may be obtained. This may be true today, but it will be a matter of only a very short time when power companies will have boosting stations in as large numbers as could be desired and there will also be garages with boosting facilities when the demand is present. If careful study is given to the transportation problem, however, boosting will not be required in commercial service except where it has been specifically allowed for in the design of the vehicle.

A. M. Pearson.[1] The principal feature which Mr. Metz's paper indicates is the fact that the horse is a thing of the past. It does not take a long stretch of imagination to see why this is so when you take into consideration the fact that the volume of business of this country has increased about 165 per cent in the last ten years, and that the number of horses and mules has increased only about 10 per cent; therefore, the only means by which this increased business, which means increased distance, as well as increased volume could be overcome, was to substitute a mechanical method of transportation for a physical method. But there are still many serious problems facing truck manufacturers and truck distributors, the largest one of which is the one called the customers' problem; that is,

[1]The Locomobile Company of America, 2314 Market St., Philadelphia, Pa.

methods by which the short haul and the facilitating of loading and unloading of merchandise, and the increase of the efficiency of a single unit in the form of a truck may be accomplished.

There are still many things to be desired. For instance, the company that I represent makes a truck with an engine which has a capacity of 45 h.p. It is obvious that there should never be a state of affairs where a truck with such an engine would stand idle while six or eight men load it. The power of this engine should be utilized in every instance for loading and unloading, and all the concomitant features of transportation to bring into existence the full efficiency of a motor vehicle.

We know how to build a chassis; we know how to build engines, both electric and gasolene; but there are many other things which are directly in line with the craft which this Society represents that are badly needed and clearly indicated. For instance, we need folding boxes, and we need them very badly. There have been many folding boxes placed on the market, but there seems to be some weakness in the hinges, which prevents the article from being a practical part of transportation.

We need different types of dumping bodies from those that are now offered to the public by body makers. Too many people are trying merely to place a truck instead of so many horses, and too many body builders are offering us the same conditions as we had with horse wagons. What we need are bodies that are made and adapted especially for motor vehicles. We need different power appliances; we need a satisfactory winch for a truck; we need a satisfactory crane for a truck, and in all these appliances we need a condition whereby the number of men necessary to operate them is reduced to a minimum.

L. H. Flanders.[1] The general agreement in results and conclusions of the paper with the reports of the electrical engineering department of the Massachusetts Institute of Technology in its exhaustive investigations of The Economical Transportation of Merchandise in Metropolitan Districts, is observed.

In the last report, Vehicle Research Bulletin No. 3, presented last March before the Electric Vehicle Association, Messrs. Pender and Thompson show how important is the study of service requirements, in securing a proper selection of the size and type of vehicle to give a resultant minimum cost. Among the figures they present the time

[1]The Electric Storage Battery Co., Philadelphia, Pa.

the wheels are in motion to the time the vehicle is in service, i.e., the mileage factor, is most impressive.

Referring to Mr. Metz's paper, Table 13, the average mileage per day runs from 20 to 24 for the electrical vehicles at a cost of from 33.8 to 36.1 cents per mile, of which roughly two-thirds is labor for drivers and helpers, while with the horse-drawn vehicle the driver and helper wages amount to about 44 per cent of the cost per mile. Anything therefore that will increase the percentage of the time the wheels are turning will decrease the cost of operation per mile.

It would also appear that with a mileage factor such as would necessarily exist it would often be foolish to run even a 2000 lb. capacity vehicle continuously at the speed that now prevails. With a reduction of speed, vehicle and battery maintenance drop an amount well worth the sacrifice. To illustrate: Assume from Table 13 that the 2000-lb. electric truck is in service 10 hours. It has a maximum speed on the level somewhere near 12 miles per hour, and an average speed of say 8 miles per hour, which would mean for 20 miles per day an actual running time of 2½ hours, with an idle time of 7½ hours. By cutting the maximum speed to 9 miles per hour the time of running would be increased to only 3 hours, which would necessitate a reduction of half an hour in the idle time, or approximately 7 per cent. This is particularly true in congested city streets where it is impossible to utilize the high speed on account of frequent stopping and starting.

Effecting this change through increased gear ratio would mean reduced battery draw for a given tractive effort or a greater tractive effort for hills and greatly reduced vehicle and tire depreciation. To the very fact that the electric vehicle is a so-called slow-speed machine may be attributed one of the reasons for its superior economy in nearly every situation over the commercial gasolene vehicle. The larger the scale upon which the electric vehicle is used the greater is the economy, particularly in battery efficiency and maintenance.

The radius of operation with electric vehicles has been much increased in the last few years by coöperation between the vehicle manufacturer, the motor manufacturer and the battery manufacturer, each designing his part to make a harmonious equipment suited to the particular service.

Attention is called in the abstract of the paper to the larger radius of action of the gasolene machine as well as to its greater cost of operation as compared with the electric car. In this connection there

would appear to be many an opportunity for increasing the radius of action of the electric vehicle by changing batteries or boosting during periods of idleness at outlying distribution points, or at the home loading platform during the noon hour.

In many delivery systems it is the practice to carry merchandise to outlying substations in large gasolene transfer trucks and from the substation to the points of ultimate destination in electric delivery wagons. In collection service the process is reversed. Many of these substations readily lend themselves to taking care of large electric transfer trucks by allowing for changing batteries which will take from 20 to 30 minutes, or by providing facilities for boosting charges. By changing batteries the radius of action of such trucks is, of course, doubled.

The lead storage battery, particularly with that form of construction using protected cores, is adapted to be recharged at exceedingly high rates and with remarkable efficiency for short periods. To be specific, that form may be recharged at an energy efficiency of over 84 per cent and a current efficiency of upwards of 97 per cent. The battery may be recharged at a rate in amperes equivalent to its state of discharge in ampere-hours. This would, of course, mean a decreasing rate of charge. This can be secured practically by providing a constant potential circuit of approximately 2.3 volts per cell, and simply by plugging in the battery without intervening resistance. By this method of charge with 100 per cent of the battery discharge 23 per cent can be restored in 20 minutes; 32½ per cent can be restored in 30 minutes; 52½ per cent can be restored in 1 hour.

With the battery three-quarters discharged 44 per cent of the total capacity can be restored in one hour, giving a total capacity of 144 per cent of the normal capacity on one charge and a corresponding increase of mileage per charge.

Where constant potential circuits of suitable voltage are not available, the battery may be boosted at a constant current, provided gassing (which means wasted energy), and high temperatures are not produced. A safe rule within this limit is to charge at a current rate in amperes equal to the discharge state of the battery in ampere hours divided by the time in hours available for charging, plus one. To illustrate: The ampere-hour meter shows 100 ampere-hours have been discharged from the battery; one hour is available for boosting; the

charging rate will be $\frac{100}{1+1} = 50$ amperes for one hour. If 15 minutes are available for charging the rate would be $\frac{100}{1+0.25} = 80$ amperes and the input would therefore be 20 ampere-hours.

Referring to Table 11, this figure would mean on the basis of 3.69 ampere-hours consumption per mile with 95 per cent current efficiency, about 13 miles increased action with one hour available for boosting.

An ampere-hour meter would be necessary as a guide in following this method, and in general the results would be as follows: A vehicle after having given 40 miles on a charge could be boosted so as to give additional mileage as follows: in 20 minutes, 10 miles; 40 minutes, 16 miles additional; 60 minutes, 20 miles additional; 80 minutes, 22.8 miles additional.

The same vehicle could be boosted for one hour after having given 10, 20 and 30 miles from a fully charged battery so as to give 5, 10 and 15 miles respectively.

These data show the possibility for long hauls and the flexibility of application of the electric vehicle.

William P. Kennedy. There is general demand for operation cost information such as is given by Mr. Metz. There has been considerable difficulty in getting accurate cost information from users of motor vehicles, particularly for the reason which is illustrated in this paper, that the installations are made partially instead of completely, and with the partial installation retaining part of the former horse installation, it is hard to segregate cost to do justice to the motor vehicle. Mr. Metz's paper has taken care of this very nicely and shown differences in cost as to partial, initial and final installation. The best practice would be completely to analyze the situation first and prepare a statement of the operating costs, and to use that as a standard table, so to speak, by which the entire proposition might be worked out. It is not just nor desirable to make comparison of unit costs of large motor equipment as against the cost of the horse-drawn vehicle in the horse-equipment installation; the only way to arrive at cost in either horse installation or motor installation is to take the entire cost of operation, preferably on an annual basis and, with all the charges against the installation, determine the operating unit cost.

John Younger.[1] In the abstract of the paper there is a statement rather damaging to the gasolene truck, and not at all substantiated by the figures given. Many like statements are made by those favoring the electric storage battery truck, but when the test of actual accounting is applied, they are found to be without actual foundation in fact.

The only gasolene cars mentioned in Par. 8 are the Cadillac, Brush, Buick, Ford, Franklin and Maxwell. These are distinctly not commercial vehicles, as properly understood, but merely touring cars, probably specially adapted. These cars run at high speed and it is a well-known fact that it is exceedingly difficult to keep the drivers from abusing them by stolen "joy rides" and by fast driving over bad roads. For these and many other reasons, their cost of operation is on the high side, but should not be as high as stated in Table 5 if there had been careful supervision. For instance, the running sheet of a gasolene 5-ton truck selected at random shows that it ran at a cost per mile of $0.267, i.e., exactly twice the cost of running the so-called gasolene commercial car in Mr. Metz's paper.

It may be interesting to state here the method used by the Pierce-Arrow Motor Car Company in taking operating costs of their 5-ton truck. It will be noticed that every item of importance in determining a true and accurate cost is taken into account, including insurance and garage charges, and a reasonable interest at 6 per cent per annum on the investment. Mr. Metz's figure of 2 per cent on Table 12 is altogether too low, though, of course, in agreement with that allowed on his horse-drawn vehicles.

| | |
|---|---|
| Investment of truck with full equipment, including body and all tools | $4800.00 |
| Interest at 6 per cent per annum | $288.00 |
| General insurance | 200.00 |
| Garage at $30 per month | 360.00 |
| Driver at $21 per week | 1092.00 |
| Fixed charges per year | $1940.00 |
| Fixed charges per day (1/365) | 5.32 |

RUNNING COSTS PER MILE

| | |
|---|---|
| Tires (8000 miles guaranteed) at $468[2] | $0.05850 |
| Gasolene, 4½ mi. per gal. at 18 cents per gal | 0.04000 |
| Lubrication | |
| Motor, 250 mi. per gal. at 60 cents per gal | 0.00240 |

[1]Mechanical Engineer, Truck Department, The Pierce-Arrow Motor Car Co., Buffalo, N. Y.

[2]Based on cost at present time.

| | |
|---|---|
| Transmission, 5000 mi. per 5 gal. at 65 cents | 0.00065 |
| Rear Axle, 5000 mi. per 5 gal. at 65 cents | 0.00065 |
| Allowance for daily repairs and overhaul (maintained and overhauled at 15,000 miles) | 0.02500 |
| Depreciation at 150,000 miles, or seven years | 0.03200 |
| Operating expenses for 1 mile | $0.15820 |

These figures are based on actual average working and can be considerably improved on by careful supervision. Tire mileage, for instance, can be bettered by the driver exercising precaution on bad roads.

We find that the field of the gasolene truck is limited only by the length of the highways. We have 5-ton trucks operating successfully in short hauls as well as long hauls. There is a very good description of a short haul job in the Engineering Record, October 26, 1912.

In our experience a comparison of costs with horse haulage is rarely asked for nowadays. There are many other benefits accruing beyond monetary saving. Users, for instance, are able to extend their territory and push sales at greater distance. Other users find the saving in time a valuable item. Still other users are operating trucks successfully where horses and mules are practically impossible. The extremes of heat and cold in our climate have rendered the horse an "unstable" animal, and the motor truck, electric or gasolene, is surely replacing him.

E. R. Gurney. In Table 8, the operating expense of a two-horse team is given as $1559.24 per year, which, reduced to a ton-mile basis, assuming 12 miles per day, 200 days, loaded one way, gives 33 cents per ton mile. This figure checks very closely with an average of horse-drawn costs as we have been able to compile it.

In Table 12, the operating expense of a 5000-lb. electric truck reduced to a ton-mile basis is 27 cents per ton mile assuming 24 miles per day 300 days. This is within 1 cent of the result obtained as an average of our 6000-lb. gasolene truck.

We have no data on electric operation nor on the lighter gasolene delivery systems. Our data begin where this paper leaves off and deal with the heavier class of units reduced to a cost per ton-mile basis.

Harrington Emerson. Reliable and adequate comparative records of costs of motor trucks and horse-drawn vehicles at the present time would be exceedingly valuable. The conclusions reached are not satisfactory, however, as the data are not standard. For instance, in

estimating the cost of a truck Mr. Metz puts down interest at 2 per cent per annum which is not a legitimate industrial charge. Six per cent, perhaps 7 per cent, and in certain portions of the country even higher, would be the very lowest that ought to be charged. Depreciation is put down at 10 per cent. There is inadequate experience to justify the assumption that the depreciation of an electric motor truck is only 10 per cent per annum. In ordinary machinery we assume about 10 per cent a year for depreciation, and certainly on such an article as a motor truck, 10 per cent is too low for depreciation. On the other hand, when it comes to the horse-drawn vehicle, we have the statement that five carts, five horses and five men were displaced. How do we know that three men could not have done the work perfectly and three horses and three carts? Usually these horse-drawn vehicles are not operated at more than 60 per cent of possible rational efficiency. It is absolutely impossible to compare a truck operating probably at 100 per cent efficiency and a horse-drawn vehicle whose efficiency we do not know. In one table the horse-drawn vehicle is put down as costing \$1.92 a day and in the very next table the cost is put down at \$1.55, showing no standard of costs. What evidence is there that a horse-drawn vehicle ought to cost \$1.92 a day? If the \$1.92 is not correct, and is merely a record of incidental cost instead of in any way being a standard cost, all conclusions drawn from the figures are invalidated.

Charles W. Baker. The figures on depreciation seem open to criticism. Mr. Metz allows 15 per cent depreciation on the horse-drawn commercial vehicle, but I know of such vehicles that have been used for many years and are not yet worn out. He charges the same rate of depreciation on the motor vehicle, but my observation is that the depreciation of such vehicles is very rapid indeed.

The Author. Mr. Flanders recommends a reduction in speed, giving as his reason therefor, that it would greatly reduce vehicle and tire depreciation. While it is true that the speed reduction would reduce this depreciation, it should also be remembered that a necessary reduction in speed in congested city streets makes it imperative to obtain a higher speed at certain times in order to get mileage out of the cars sufficient to make them pay. The exact speed at which commercial vehicles should be run is open to question, but I believe that the present speed is about as nearly right as it can be made under present conditions.

Referring to the discussion by Mr. Emerson, it is my opinion that interest charges should be made to agree with the actual interest which any particular establishment has to pay. If the government can borrow money at 2 per cent, it is fair to charge 2 per cent interest on investment for government work. If private establishments have to pay, say 4 or 5 per cent, then they should charge 4 or 5 per cent interest on their investment.

The figures in my paper were given from actual experience, and the charges were therefore made to agree with the facts in the case.

It will be an easy matter for anyone desiring to install automobiles to separate the interest charges and make his particular interest charge agree with the interest which he will have to pay.

Referring to the depreciation charges, it is a well-known fact that the horse is not much good for heavy trucking service after five years of use, and the horse-drawn wagon is not much good for the same service after ten years of use, so that the 20 per cent depreciation on the horse, 10 per cent on the wagon, and 15 per cent on the harness represent average figures. The depreciation on electrics is, of course, not yet well understood, but there are plenty of electric vehicles in use at the present time which were purchased ten years ago, or more, so that the 10 per cent depreciation for this class of vehicles is on the safe side.

No. 1392

# PRACTICAL OPERATION OF GAS ENGINES USING BLAST-FURNACE GAS AS FUEL

By Charles C. Sampson, Joliet, Ill.

Member of the Society

The question of the operation of gas engines using blast-furnace gas as fuel includes several important factors outside the actual operation of the engines themselves. It will therefore be aside from the present purpose to do more than state that the usual blast-furnace gas has the following composition:

AVERAGE OF DAILY ANALYSIS FOR SIX MONTHS

| | Per Cent of Dry Gas |
|---|---|
| Carbon monoxide | 27.1 |
| Carbon dioxide | 121 |
| Hydrogen | 3.1 |
| Methane | 0.1 |
| Vapor of water from 3 to 5 grains per ft. dry gas | |
| Nitrogen | 57.6 |

and a calorific or heating value of about 97.6 B.t.u. per cu. ft., and gives a consumption of 95 cu. ft. per i.h.p-hr. in the engine.

## CLEANING OF THE GAS

2 One of the most important of these factors and one which held back the general use of these engines many years is the cleaning of the gas. As delivered by the furnaces to the downcomer the gas contains normally from 3 to 10 grains of dust per cubic foot of dry gas, but at times of slips or other sudden changes in the furnace. it carries much more. For use in engines the gas must be cleaned at most to 0.02 grains of dust to satisfy the requirements of the engine builders, but even this figure is too high to satisfy the operating engineer since it is possible to clean the gas to 0.005 or 0.006 grains per cu. ft. with great benefit to the engines.

3 The method of cleaning most used at present has three stages: (*a*) dry cleaning to 1½ to 2 grains per cu. ft. which is always done

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

by the blast-furnace department; (*b*) primary washing in static washers to about 0.15 grains per cu. ft.; (*c*) dynamic or mechanical cleaning in highly developed machines to 0.015 or less. The last stages are usually handled by the gas-engine departments, though as the furnace men realize more and more that a cleanliness of 0.2 grains per cu. ft. or less is of great benefit to the stoves and boilers they will take over the second stage, leaving only the final cleaning for the gas-engine department.

4 The dry cleaning is done in dry dust catchers, the standard design being a large diameter, vertical, cylindrical shell into which the gas enters tangentially near the top and leaves through a vertical outlet pipe which extends about two-thirds down from the top. These dust catchers remove the heavier particles of dust, but their efficiency is only about 80 per cent as they pick up, or perhaps do not drop, the finer dust which is carried on by the upward current of gas to the outlet.

5 The refinement of design in dry cleaners has advanced materially in the past three or four years, as shown in the modern apparatus resulting from the careful study of the problem. One of the latest of these is the centrifugal dust catcher shown in Fig. 1. This device makes use of the centrifugal separation of dust from the gas as it passes inward through a cylindrical spiral opening into a dust basin at the bottom. The gas enters at the top of the outside, leaves at the top of the inner end of the spiral and passes upward through an extension of the pipe around which it is wrapped. The gas passes free of all obstructions at the upper end of the spiral while the dust separated drops to the bottom through the open end. There is no tendency for the gas to pick up the separated dust and carry it out as is the case in the older types of dry cleaners.

6 It is frequently found that sudden changes in the direction of flow of the gas, as at water seals or other necessary bends in the pipe, are quite efficient in the removal of dust. In one case gas carrying about 5 grains per cu. ft. passed through four sharp bends and gave all dust but about 2 grains per cu. ft. For this reason every part of the dry gas main where such bends are necessary can be made to assist materially in the cleaning of the gas, if pockets are added equipped with valves so that the dust can be conveniently removed.

7 Where long gas mains are necessary they can be made to add to the cleaning of the gas by building them in successive lengths with sufficient rise and fall to allow the dust to settle in pockets at the bottom angles for cleaning. If the gas for any reason moves slowly

in a long main the loss of heat through the pipe will probably reduce the temperature below the dew point and thus condense some of the moisture carried with the gas from the furnace and cause the deposit of wet dust which adds greatly to the cleaning plant labor. This is especially apt to occur where two or more groups of furnaces supply one washing plant; the gas from the one with the lower top pressure will move slowly or even reverse its direction of flow at times, allowing excessive cooling and the resulting condensation. This condensa-

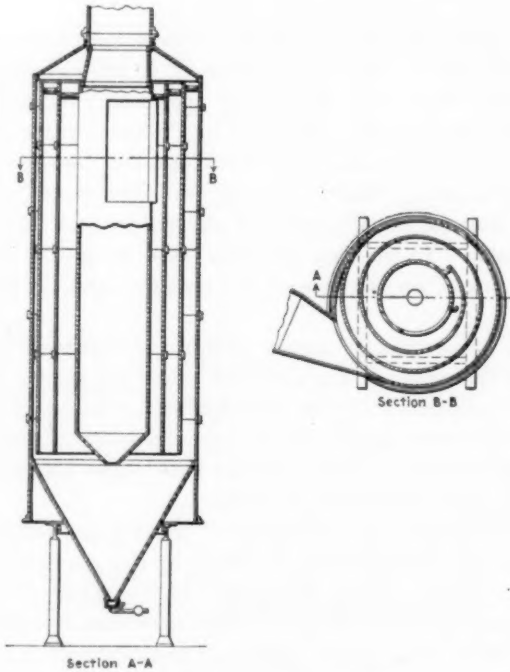



FIG. 1 CENTRIFUGAL DUST CATCHER

tion will begin when the temperature is reduced to 115 deg. to 120 deg. fahr. and will of course give more trouble in winter when the condensed moisture will freeze in the dust valves and drips and require continual thawing to allow its removal.

8 It is possible to keep the gas mains clean without taking them out of service if they are equipped with sufficient openings to allow every part of the pipe to be reached with a stream from a high-pressure water system, and with valves or doors at all low

points for the removal of the mud washed down. The mains near the furnace of course do not need this equipment as they can easily be designed to be entirely self-cleaning, while it is quite necessary that long mains where condensation may occur be so equipped.

9 The present primary washers (the first stage of wet cleaning) are of the static scrubber type and include all those in which the gas passes through a stationary shell without moving parts, the water for washing being supplied either in spray or sheets. The spray and hurdle, Mullen, baffle, and rain type scrubbers come under this classification.

10 The spray and hurdle system is preferred on account of its better distribution of water, and since it is self-cleaning it needs inspection only after long periods of operation. Several of these scrubbers have been opened after from one to three years' service and in every case have been found perfectly clean and required no repairs whatever before being returned to service. The wood was in good condition as it is continually wet and oxygen does not have access to it to start decay. In the rain or baffle types the gas is more apt to channel and travel up one side of the scrubber and the water down the other.

11 It is important to secure uniform distribution of the gas as well as of the water in any scrubber. For the inlet a cone about two-thirds the diameter of the shell with a cone-shaped ring below it open in the center about one-half the diameter of the shell will give good distribution. These should both slope about 45 deg. to keep the mud from remaining on them.

12 Two outlets at opposite sides of the top are better than one on account of the deflection of the water by the gas currents if only one is used. This is particularly true if the water is sprayed by falling on spray plates as the gas current may then be strong enough to blow the water clear of the plate and thus entirely lose its effect. Spray nozzles are not subject to this fault but are not able to handle water that has much dirt in it without a great amount of attention.

13 In designing the scrubber bottom, its foundation and the basin and overflow for the outlet water, it must be remembered that while the usual working pressure will be from 6 in. to 18 in. of water, a slip will give pressure of from 40 in. to 50 in. for a short time. A normal head of water of 36 in. from the bottom of the scrubber to the water overflow level with the basin walls 24 in. above this and an emergency overflow 4 in. below the top of the basin walls will care for slip pressure without blowing out any gas or overflowing

the basin into the yard. The bottom of the basin wall will be self-cleaning if it has a steep slope and the outlet pipe is from the center of the bottom. The whole design of scrubber and basin must be examined to eliminate all places where mud can remain long enough to cake. Fig. 2 shows this arrangement of scrubber bottom.

14 Should the water overflow pipe be stopped even for a short

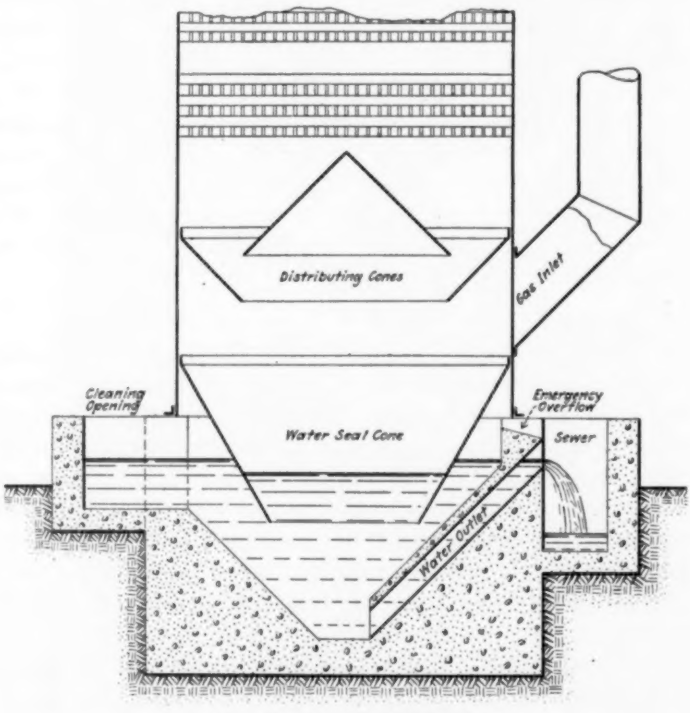



FIG. 2 SCRUBBER BOTTOM

time the heavy mud will settle to the bottom of the basin and when the overflow pipe is cleaned there will be such a quantity that even the extra head of water to the emergency overflow will not force it out. For this reason the forming of heavy chunks must be prevented as much as possible and provision must be made for stirring the basin water both with hoes or rakes and with a stream of water from the end of a pipe which can be thrust into all parts of it. It will be found convenient also to have the pipe bent at the end so that the stream can be directed up the overflow pipe to furnish additional head

for starting the flow when necessary, or a special pipe with return bend and short nipple to thrust down the overflow pipe itself will surely be able to start the flow.

15 The final stage in cleaning is done with mechanical scrubbers or washers. These are highly developed and the Theisen patented gas washer has been in the lead for several years though other types are now being worked out, their builders claiming better results with less water and power consumption than the Theisen. The Theisen washers require about 3 per cent of the power-plant output for their operation and from 16 to 18 gal. of water per 1000 cu. ft. of gas cleaned, which added to the 75 to 80 gal. required in the scrubbers makes the total from 90 to 100 gal. for the whole cleaning process. The newer apparatus, which are along the lines of the mechanical disintegrator, claim to use about 20 gal. of water per 1000 cu. ft. of gas for the whole cleaning process and to operate on less power than the Theisen washers.

### HOLDERS

16 In blast-furnace gas-engine plants the engines are entirely dependent upon the continuous supply of gas from the furnaces; a 100,000-cu. ft. capacity holder can only be considered a pressure regulator with capacity for enough gas to allow retiring in good order when the gas supply is cut off for any reason. Thus in a 1000-kw. plant with such a holder the gas on hand would operate the plant only for about 25 to 30 minutes and should not be counted on for more than 15 to 20 minutes. This in an emergency would give time to notify the various departments using power and allow them time to prepare for a shutdown.

17 The quantity of gas consumed by the engines is regulated by the governor to suit the power output, but since they must be supplied with gas at uniform pressure for satisfactory operation, it is necessary to regulate the gas supply by some type of gasometer. This is best done by a gasometer of capacity such that the pressure fluctuations are not noticeable at the engines, and since it is well to have an emergency quantity of gas the gas holder itself will meet both demands at once if supplied with an efficient regulation valve. The holder will regulate the pressure perfectly between the maximum limit of the total quantity of gas that can be forced through the mains with the furnace pressure available assisted by the gas washers and the minimum limit of the leakage at the regulating valve.

18 There should also be the possibility of regulating the gas

quantity at the secondary washers since at times of very light loads the gas pressure between the holder and washer may blow out drip seals or cause dangerous gas leaks. This can be cared for by the installation of butterfly valves with quadrants either before or after the mechanical washers. The latter is to be preferred for then the gas remains longer in the washers and receives additional cleaning.

19 A good regulating valve at the holder is a butterfly valve attached by means of levers and cables to the holder bell so that it will remain wide open until the bell rises within a few feet of its

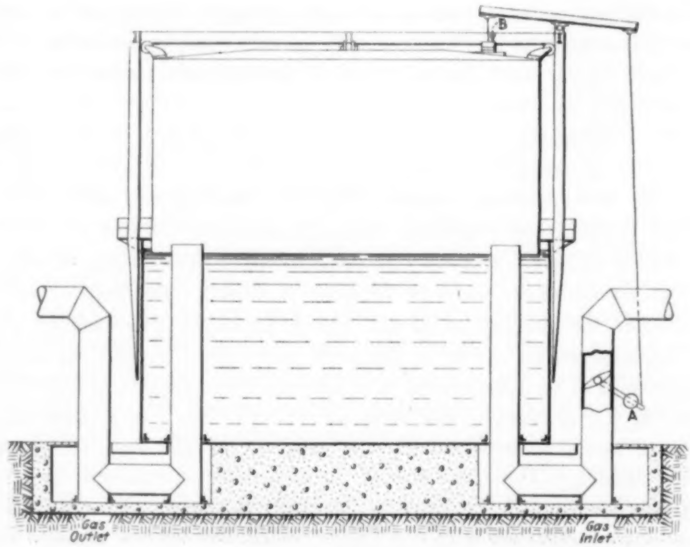



Fig. 3 Arrangement of Regulating Valve for Gas-Holder

upper position, and close gradually till at the highest position it is completely closed. The arrangement shown in Fig. 3 works satisfactorily. The weight *A* must be heavy enough to close the valve and the weight *B* must be heavy enough to open the valve and also lift the weight *A*.

20 All exposed water lines must be protected from freezing. This is especially true of the supply to seals, drips from the gas main, and any line that does not have a continuous flow. With good water separators after the secondary cleaning apparatus, freezing weather or even 8 deg. or 10 deg. fahr. below zero, will not cause trouble in the

gas mains themselves, though any valves which may be nearly closed or which are closed part of the time must be carefully protected. The butterfly valve for regulating the gas should be enclosed in a tight box with steam coils to keep it in working order. This also is true of the valves at the gas-washing plant unless it is possible to install them within a heated building.

21 The water in the gas holder must also be warmed. The exhaust from the regulating valve coil will easily keep the holder water warm enough to prevent freezing except in the coldest weather (under 0 deg. fahr.), when it is usually necessary to supply additional steam through several nozzles arranged to set up a circulation of the water around the tank. These should be well down in the water or ice will form on the lower part of the shell and build in toward the center and prevent the lowering of the bell.

22 During the time the holder water is warmed it is important frequently to observe its temperature; if too hot it will charge the gas with water so that condensation and freezing will take place in the gas-engine supply pipes. When the water circulates properly in the holder it is not necessary to have it any warmer than 38 deg. or 40 deg. fahr., while a rise to 65 deg. or 70 deg. will give trouble.

23 If the gas holder is not visible from the gas-washing plant the operator needs a visible signal to give him its position, also an audible signal to inform him if it should lower beyond safe working position, the amount of the drop allowable before the audible signal operates being determined by the position of the regulating valve at the holder. The drop should be less than an amount to give a complete opening of the valve. The gas-washer operator should have telephone connections with the engine room, besides the usual whistle or bell signals which are used to notify him of the starting or stopping of engines. He should also be in close touch with the blast-furnace department in order that any change in the gas supply can be known in advance.

### RELIEF DOORS

24 In all gas-pipe lines so-called explosion doors are installed. These are as a rule useful only for access to the main for cleaning, usually being made of cast iron and hinged; on account of their weight and method of attachment the moment of inertia is so great that they will not open quickly enough to prevent the destruction of the main in which they are installed. Any gas main that will support itself over the span usually employed will easily stand any

pressure that can be produced in the cleaning plant, and the use of these valves or other relief valves is not necessary. The inconvenience of escaping gas makes it advisable to design them as cleaning doors only, and to arrange them with a clamp fastening to avoid this inconvenience. If it is thought necessary to install explosion doors or valves I would suggest the use of sheets of light material arranged in frames so that they will be blown out should an explosion occur in the main.

25 The best protection against explosions is careful operation, especially to guard against a reduction of pressure of gas at the furnace side of the cleaning plant due to no air being drawn into the main at the stoves, and to see that no piece of apparatus is put in service with air trapped so it can be mixed with the gas and sent along to the engines.

### RECORDING INSTRUMENTS

26 Thermometers and pressure gages for indicating the temperature and pressure of the gas: entering the cleaning plant, between the primary and secondary washers leaving the latter, and before and after the gas holder, form important parts of the gas-cleaning system. The ordinary gas works thermometer with a stem reaching about 8 to 10 in. into the gas mains are to be located at each of the above points, while pressure gages of the U-tube type with inches of water as a measure of the pressure can be located in the gas-washer building and connected to these points by $\frac{3}{8}$-in. or $\frac{1}{2}$-in. gas pipe. Recording gages should be used in connection with the indicating water column for the gas pressure at the entrance to the cleaning plant in the gas main leading to the gas holder.

### LOG RECORDS

27 The successful operation of the gas-washing plant is very much advanced by the proper understanding of the meaning of the variation shown by these thermometers and gages. For this reason it is important to keep a record of these variations on carefully designed daily log sheets with at least eight daily notations. The gas-washer operator soon learns to interpret the gage and thermometer changes, and will often fortell serious trouble by such understanding. For instance, the partial filling of a water seal is indicated some time before it will cause trouble by the swinging of the water in the U-tube, this movement being so markedly different from

any other that he knows at once the trouble and from the location of the gage can easily tell which seal is filling.

28 The daily log sheets should have space reserved for the operator to note any unusual occurrence and the work done to keep the plant in condition. It should be in fact a complete report of each day's work to the engineer-in-charge, keeping him in close touch with the changing conditions in the gas-cleaning system.

## ENGINE STARTING

29 A second most important factor in successful gas-engine operation is good engine operators, and the same characteristics which are valuable in steam-engine operators are valuable in the gas engineer.

30 The operation of the engines themselves is exactly similar as far as the running gear is concerned and it is only the fact that the gas engineer is fireman as well as engineer that makes it necessary that he be more alert and watchful. Economical operation of gas engines on the same account requires that the engine operator must have his sense of "the feel of the machine" well developed.

31 Compressed air at from 150 to 200 lb. per sq. in. pressure has proved satisfactory for starting gas engines and is especially desirable on account of the ease with which a suitable quantity can be stored under pressure ready for use at any time.

32 In a starting system of 2000 cu. ft. capacity the air pressure is lowered about 20 lb. in starting one 3000-kw. twin-tandem unit, and since 150 lb. pressure is sufficient for a start there is a possibility of at least three starts from 200 lb. initial pressure, which is certainly sufficient to get under way even during the excitement of an emergency shutdown.

33 Record was kept of the pressure drop in starting an 1800-h.p. twin-tandem Allis-Chalmers engine from an air system having two tanks of 1100 cu. ft. capacity each. This record included 19 starts, 16 using the full capacity of the system and 3 with one tank out of service. This record is plotted in Fig. 4, the pressure in the system being shown as ordinates and the pressure drop as abscissae; the 16 starts with complete air system in use are indicated by circles, and the 3 that were made with one of the air tanks shut off by crosses. It may be noted that the quantity of air required to start the engine was about the same, regardless of the pressure in the air tanks.

34 The necessary capacity of air tanks and air compressors for

a given plant depends upon the number and size of engine units, and the frequency with which they may need to be started. After the engine operator becomes familiar with the operating peculiarities of the engines he should be able to start them at intervals of from 4 to 8 minutes and not lower the air pressure more than the compressor can make up in that time, if an engine lowers the pressure 8 lb. per sq. in. in a 2000-cu. ft. capacity system, the compressors should compress 8/15 of 2000 = 1060 cu. ft. free air in the maximum time allowable between starts or say 10 minutes. This would require two 106-cu. ft. compressors.

35 For the ordinary blast-furnace gas-engine plant of from three

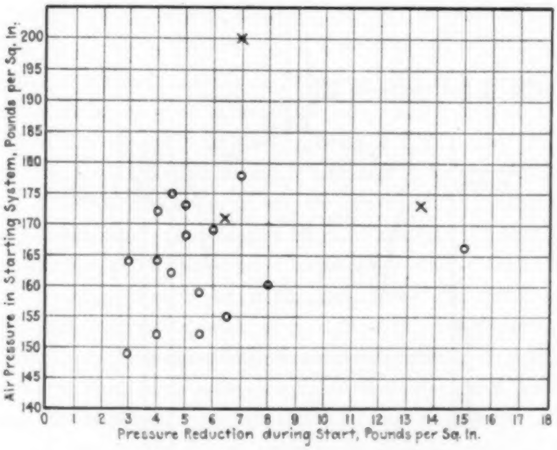



FIG. 4 RECORD OF PRESSURE DROP

to six engines two air compressors of 100 cu. ft. capacity and air tank capacity of 2000 cu. ft. are quite sufficient, while for more than six engines the compressor capacity should be increased rather than the tank volume. At least one of the compressors must derive its power from some source outside of the gas engine in order to be able to start the plant if all units should be down.

36 It is important to keep the water jackets thoroughly clean and the item of jacket cleaning should appear regularly in the engine operation schedule. This cleaning requires careful attention since, with the class of labor usually put on this work, it will be slighted in the places where the most care is needed.

LUBRICATION, CLEANING OIL

37 The question of lubrication is one of so many variations I can only say that for general lubrication of such as main bearings, crosshead and crankpins and crosshead slides where the rubbing surfaces are at room temperature, an oil of the following physical characteristics has given excellent service:

| | |
|---|---|
| Specific gravity | .888 |
| Viscosity (Tagliabue) | 210 at 70 deg. fahr. |
| Cold test | 35 deg. fahr. |
| Flash temperature | 435 |

38 This service also includes satisfactory separation of water and dirt by settling and filtration. On account of the almost certain

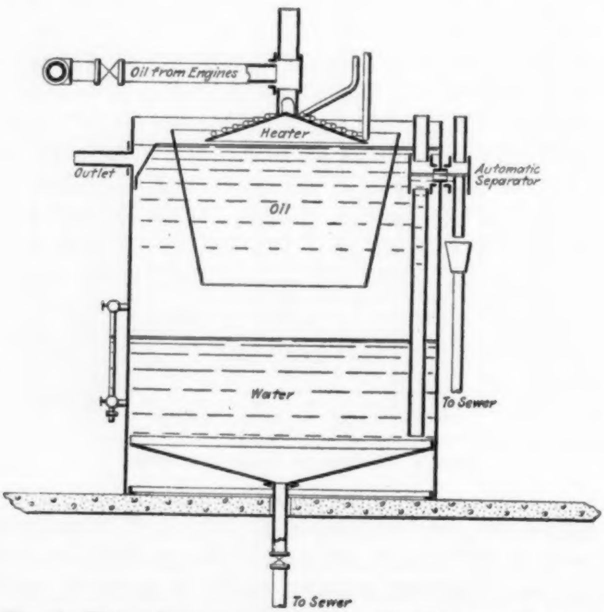



FIG. 5 WATER-SEPARATING TANK FOR OIL-CLEANING SYSTEM

mixing of water from the cooling system with the system oil, it is necessary to provide means of separating the water and oil in the filtration process and it can be done thoroughly only by heating the oil to about 160 deg. or 190 deg. fahr. and giving it time in a quiet condition to allow the separation. A large part of the dirt will settle with the water. Such that does not, must be removed by filtration

through fine cloth either of organic fiber or of fine wire. The latter is more to be desired because of the ease with which it can be cleaned.

39 A good oil-cleaning system giving excellent satisfaction consists of one 1500-gal. water-separating tank, shown in Fig. 5, with a heating coil over which the oil flows as it enters on returning from the engines, and an adjustable automatic water overflow to discharge the separated water, two settling tanks of the same size through which the oil passes in tandem to allow time for quiet settling of dirt particles, and a filter unit with 20 filter bags, 10 each in two filter tanks.

40 An extra tank is used when either of the other three is out of service for cleaning. An auxiliary tank of about 200 gal. capacity is used for "boiling up" the sludge taken from either of the large tanks or the filters at time of cleaning as well as such dirty oil as can be drawn off daily from the bottom of the overhead oil tank.

41 This system is shown in Fig. 6. The oil from the engine drips enters tank *A* over the steam coil, flows down through the inner cone, then up and out the overflow to *C* and *D*, thence to the filters *F* and *G* through E, which is also a water separator. The clean oil is pumped from the filters by one of the duplicate pumps at *K*, to the overhead engine supply tank in which the quantity of oil on hand is shown by an index on a large gage visible from the engine-room floor. Gage glasses on each tank show the level of the line between the oil and water both as an operating convenience and as a means of checking the quantity of oil used during the month. The separated water flows down the inside of the cone in *A* to the bottom of the tank from which is flows through the automatic overflow *H*. The nipple in the tee at *H* is adjustable so that the water in *A* can be held at the level found best in operation.

42 A part of the dirt is oil-coated so that it floats between the water and the oil and will accumulate until its removal is necessary. The oil from the engines is then turned into tank *B*, the supply to *C* and *D* being kept up by stopping the water overflow and filling *A* with water as long as good oil flows out. The water is then drawn off to the sewer and the sludge pumped into the boiling tank *J* where as much oil is reclaimed as possible. The other tanks are cleaned in the same way. There are pipe connections from the bottom of all tanks to one of the pumps, also from the discharge of this pump to the tank *J*.

43 Such an oil system will keep the oil clean for a plant circulating 500 to 600 gal. per hour. Of course some oil is lost through

leakage at the engines, and some is wiped up in keeping the engines clean, but the addition of new oil need not amount to more than 100 gal. per month. In blowing-engine plants where the engine oil is drawn into the blowing cylinders from mechanically operated valves the oil consumption will not be so low unless good oil separators are installed in the cold blast mains arranged to discharge this oil back into the oil system.

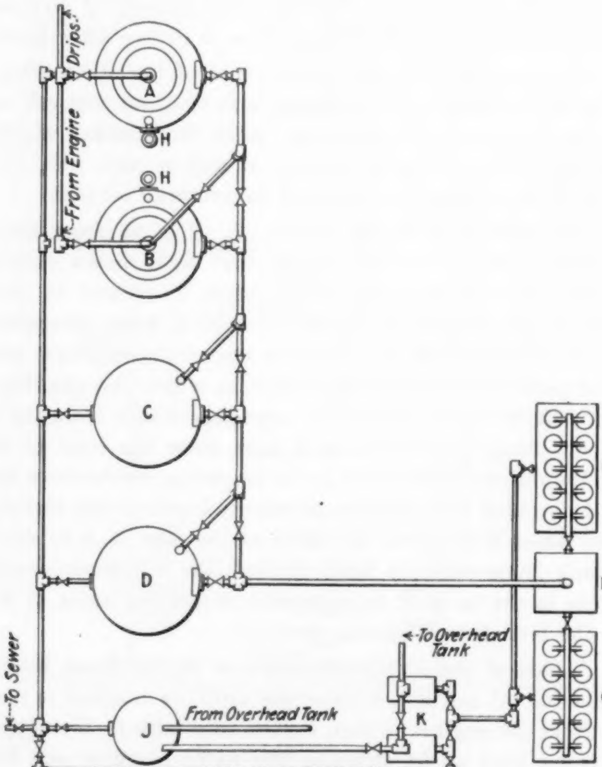



Fig. 6 Oil-Cleaning System: *A* and *B*, Water-Separating Tanks; *F* and *G* Filters; *J* Boiling Tank

44 The cylinder oil question is also one of many opinions. The varying cleanliness of the gas, hardness of cylinder walls and piston rings, piston speeds, mean effective and maximum pressures all have their influence on the action of the cylinder oil. An oil showing a specific gravity of 0.902; viscosity (Tagliabue) of 78 at

212 deg. fahr., and a flash temperature of 380, gave excellent results in a gas blowing-engine plant where the dust was low (0.01 or less) and piston speeds less than 600 ft. per min., and was not satisfactory in another, with 0.012 dust and piston speeds of 850 ft. per min. In the latter case the oil was replaced by one of specific gravity, 0.920; viscosity, 203 at 212 deg. fahr.; and flash temperature, 502, and immediate improvement was shown.

45 With the lighter oil the cylinders were not dry in any part though they did show more wear than was expected for the time in service, the machining marks in the bore being almost invisible after three months' operation.

46 The cylinder oil can be put in a tank in the basement and piped to all cylinder oil pumps by using compressed air at 15 lb. per sq. in. This provides opportunity for the installation of oil meters to keep accurate account of the oil used on each engine, or the supply tank may be equipped with graduated glass and record kept of the supply to the whole plant.

### IGNITION

47 The mechanically operated igniter is much to be preferred to the magnetic type. The current supply should be from a source not liable to fluctuation, such as that from a motor generator set that supplies current to the ignition system alone and arranged in connection with a storage battery so that should anything happen to the motor generators the battery would take up the load, automatically signaling the operator. The location of the ignition plugs is important, since an explosion on one side only of the piston will force it to the other side and cause it to strike the cylinder wall. This is easily apparent in cylinders having the combustion chamber at the side and the effect of one-sided explosions can be seen when one of three equally spaced igniters is not working.

48 Premature ignition is usually caused by excess hydrogen in the gas, and will occur when the quantity of hydrogen reaches 4.6 per cent, depending also upon the cleanliness of the cylinders. This prematuring is one of the first indications of leaking cooling plates in the furnace and the gas-engine operator will often be able to inform the furnaces of this condition before they learn of it themselves. When a furnace has the wind off for casting, the water pressure in the cooling plates is greater than the furnace pressure and the water enters the furnace and is immediately dissociated; the oxygen being consumed by the coke leaves the excess hydrogen in the

gas. When the wind is put on again this gas, rich in hydrogen, is sent along to the engine causing prematuring.

MISCELLANEOUS INFORMATION

49 The pistons of the early large gas engines were of cast iron but these gave considerable trouble by cracking because they were not properly ribbed. Several builders changed to cast steel, but found they gave trouble either by cutting the cylinder walls or by beading over and binding the rings. Cast iron was again resorted to with an improved design. In the cast-steel pistons also the movement of the rings widens the grooves so that in a short time there is too much clearance which necessitates turning the grooves and making new rings.

50 Cast-iron snap rings of uniform cross-section give better service than any other type. This has been learned after many attempts to design a complicated ring, the designer believing a ring to hold gas-engine pressure would be more difficult to make than one for steam-engine pressure.

51 Piston-rod packing furnishes one of the difficult problems for the designer and also for the engine operator. A good packing must be simple in design and, as in the case with the piston rings, the forms with the fewest parts seem to give the best service. Both cast iron and babbitt have given good results. The success or failure of this part of the engine depends largely upon its cleanliness and too much care cannot be used in assembling it to insure its proper application.

52 The engine-room basement has not received the attention it deserves. Two items the designer of any power plant will do well to consider are: make the engine-room floor high enough over the basement floor to allow 6 ft. clear under all suspended pipes, and have the basement floor slope enough for rapid drainage (at least 12 in. for 100 ft.). A dry basement with plenty of head room is easily kept clean and a clean basement is a great help in keeping the whole room in good condition. Ventilation and lighting are also more easily accomplished in a high basement.

53 The question of the safety of the employee in all occupations is at present a live one and is to be considered in the engine room as well as in the rolling mill. Safety demands the elimination of all dangerous conditions by covering gears, guarding flywheels, generators and crossheads, enclosing electrical apparatus and keeping all gas

pipes tight and free from leaks. The men must be trained to watch out for their own and their fellow workmen's safety. It is just as important that the lives and limbs of the engine crew be protected as it is that the cost of electricity should be low.

54 Complete records of operation are invaluable as it is only by the study of accurate records of the actual happenings that we can hope to improve. It is not possible to trust to memory for a comparison of the results of different types or arrangements of apparatus.

55 A daily log of the various pressures and temperatures must be kept to learn whether the plant conditions are changing and to know the cause of these changes. Any unusual occurrence or the regular recurrence of repairs noted on the log sheets puts the information regarding the plant where it can be used in predicting and preparing for the future. Not only should this information be kept daily but it should be collected and averaged monthly and yearly for the comparison of month with month and year with year. Much of the engineering information is best shown graphically, for a sheet full of figures does not give a true conception of the actual conditions.

56 The original information is necessarily furnished on the daily log sheets written by the shift engineer and he should be supplied with a copy of the resulting data sheets. He is just as much interested in the power plant as is the chief engineer and this information cannot be placed anywhere to do more good than with the engine operators. The rate of progress in the gas-engine field depends entirely upon the rapidity with which engineers are able to gain understanding of this machinery and the collecting and compiling of these records is of the greatest service for this purpose.

## DISCUSSION

Fred. H. Wagner. From reports received, I find that with practically the same character of apparatus, (*a*) dry dust cleaners, (*b*) some sort of static towers like the Czchokke or Steinbardt, and (*c*) the Theisen washer, the final cleaning varies between 15 cents and 62 cents per 100,000 cu. ft. of blast-furnace gas cleaned; this being based on $10 per 1,000,000 gal. of water and 1 cent per kw-hr. The great discrepancy in these final figures has led to the question as to why such discrepancies should exist. The answer is a difficult one to give; for instance, at one blast furnace the final cleaning based on 100,000 cu. ft. of gas costs 61 cents, at another 47 cents, at another 42 cents,

at another 25 cents, at another 20 cents, at another 19 cents, and at another 15 cents.

Mr. Sampson mentioned that some new apparatus, which is along the line of mechanical disintegrators, claimed to use about 20 gal. of water per 1000 cu. ft. of gas for the whole cleaning process, and to operate on less power than the Theisen washers; also that with the present process from 90 to 100 gal. are used for the entire cleaning process.

At the last meeting of the West of Scotland Iron and Steel Institute, B. W. Head read a paper[1] giving the results of his visit on the European continent, for the purpose of examining blast-furnace conditions. After calling attention to the failure of the Scottish steel manufacturer to take advantage of the improved methods adopted in Germany, Belgium and France, he takes up the subject of washing blast-furnace gases for gas-engine purposes and points out the difference in the methods practised in the British Isles and on the Continent, mentioning especially the Feld vertical centrifugal washer for the cleaning of gas.

Besides the Theisen washer mentioned by Mr. Sampson, and the Feld washer, there is a third, the Schwarz. The last two are attracting a great deal of attention in Europe in connection with this process. They require less power than the Theisen washer, and the Feld requires considerably less power than the Schwarz.

At the blast furnaces in Donawitz, where 1,600,000 cu. ft. of blast-furnace gas is treated per hour, the total power with the Schwarz washers amounted to from 124 to 128 h.p. At the blast furnaces in Pompey, France, of exactly the same capacity, the Feld washers used from 50 to 55 h.p., the power given including the necessary power for exhausters. Unfortunately in my data on the Schwarz washers, I have not the power of the washer and the exhausters separate; however, for the Feld washers, the washer power amounted to from 15 to 20 h.p. and for the exhausters 35 h.p. This great difference in power requirements is due to the fact that the Feld washer throws absolutely no back pressure—the washer operating with an even gage on both the inlet and outlet, while the Schwarz washer throws a considerable back pressure.

In speaking of the quantity of water required for cleaning the gas, I would mention that two factors enter into this discussion: (*a*) being the cooling of the gas to a temperature which retains the heat

[1]The Journal, West of Scotland Iron & Steel Institute, vol. 19, Nos. 6 and 7, pp. 266 and 319.

units and at the same time reduces the volume of the gas to the smallest possible compass in order to avoid large gas-engine cylinders. The cooling of the gas is a thermal question, and the amount of water required is determined by the amount of water vapor which carries the heat contained in the gas. This amount of water would be necessary, no matter in what sort of washer the gas is cooled, as a certain number of heat units must be extracted, and the water cannot take up any more than its temperature will permit.

With a temperature of 500 deg. fahr. and treating 5,400,000 cu. ft. of gas per hour, with the gas coming from two furnaces, reducing this temperature to 86 deg. fahr. requires about 31 gal. of water per 1000 cu. ft. of gas. The cleaning of the gas after the water is cool, is done by the hot water from the cooling chambers, about one-quarter of the amount given above, or 7½ gal., being run into the washing chambers of the Feld washer. For this purpose the Feld washer is built in seven sections, the upper four acting as cooling chambers and the lower three as cleaning chambers. It is a known fact that if an impalpable powder be placed on a floor, and cold water is thrown on it, the water forms globules on the surface of the particles, but if hot water is thrown on the powder, it immediately mixes and forms a mud. This is the principle on which the Feld washer is operated. Each one of the sections of this washer contains a series of perforated truncated cones, the lower ends of the cones dipping into the water, and by the revolving of these cones, given a periphery speed of 1600 ft. per min., the water is carried up inside these cones and hurled out through the perforations in a finely divided spray.

Those conversant with dust washing will admit that the proper method to wash dust out of gas is to bring the gas into intimate contact with the smallest particles of water possible, and this is done in the Feld washer.

In a lead smelting furnace, where these Feld washers are in use, these washers recover six tons of dust containing from 80 to 100 per cent lead every 24 hours and entirely remove the lead particles, which formerly escaped into the atmosphere to the detriment of the surrounding vegetation.

In the blast-furnace plant at Pompey, France, where the Feld washers are also in use for cleaning blast-furnace gas for gas-engine purposes, I would state that prior to the use of the Feld washers, it was necessary to open up the valve chests on the engines about once

every ten days and remove the accumulated dust; since the installation of the Feld system, it has become necessary to open up the valve chests only about once every three months. These washers have been in continuous operation for about 30 months without one moment's shutdown for cleaning or repairs, the washers being self-cleaning; the water carrying the dust leaves the washer in the shape of mud, which is easily handled by means of centrifugal pumps.

The aim of the modern engineer is to reduce the cost of operation and at the same time to secure better results. This has led to the Feld washer, and I would earnestly recommend anyone treating furnace gases for gas-engine purposes to investigate the Feld washer before purchasing other apparatus, as the first cost of the initial installation, as well as the ground space occupied is less than with the use of static towers.

A washer of the Feld type has just been placed in the works of the American Smelting and Refining Company at Maurer, N. J., for the purpose of removing the last traces of gold, silver, lead and selenium from the gas which comes from the mud cupel furnaces.

In order to clean 100,000 cu. ft. of blast-furnace gas for gas-engine purpose, or so that the final gas does not contain more than 0.01 grains of dust per cu. ft. and to reduce the temperature from 500 deg. to 86 deg. fahr. would cost with the use of nine Steinbardt coolers with the necessary Theisen washers, less labor and exhauster power, 14½ cents; with nine Steinbardt coolers and Schwarz washers, 11 cents; and with three Feld primary and two Feld final washers 5.2 cents. This is based on the cost of water at $10 per 1,000,000 gal. and power at 1 cent per kw-hr. in a plant capable of handling 5,400,000 cu. ft. of gas per hour.

The author did not desire to present a closure.—Editor.

# SYMPOSIUM ON FIRE PROTECTION

## No. 1393 a

## DEBARMENT OF CITY CONFLAGRATIONS

By Albert Blauvelt, Chicago, Ill.

Member of the Society

Opinions take opposite sides as to whether the central districts of our leading cities are today subject to conflagration.

2 Passing over interested and private opinions, the 1911 report of the Joint Committee of the Senate and Assembly of the State of New York, on insurance and fire waste; the Illinois Fire Insurance Commission Report of 1911; the Ohio Fire Marshal Report for 1907; the United States Department of the Interior, Bulletin 418, of 1910; the Wisconsin Senate and Assembly Committee of 1913 and all other public reports the writer can find agree that each of our cities is today subject to conflagration. Our cities appear to spend in public and private ways enough money to make a conflagration impossible for the central districts, but they do not achieve this result.

3 The horizontal hot blast of a heavy fire driven by wind ranges from a few hundred to more than a thousand feet, whereas the most powerful hose streams are not effective at over one hundred to two hundred feet.

4 Such fires develop in the heart of our cities in the face of more firemen and apparatus than can be quickly used, as at Boston in 1872 and 1889; Montreal in 1901; Paterson in 1902; Baltimore and Toronto in 1904; and such fires also develop in city outskirts or cheap quarters and by force of wind sweep into the city proper as at Portland, Me., in 1866; Chicago in 1871; St. John, N. B., in 1877; Ottawa-Hull in 1900; Jacksonville, Florida, in 1901; at Chelsea, Mass., and other cities.

5 Inasmuch as any engineering plan to debar conflagration, if intended to cover an entire city, inclusive of cheap districts, would be prohibitive in cost; any such plan must be limited to the central or high value district and designed to prevent free spread of fire

---

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

within same, and also debar any deep inroad of conflagration from without such district.

6 Every conflagration must necessarily begin in one of two ways, or a combination of both, Chicago in 1871 and Baltimore in 1904 being examples of the two types.

7 The Chicago fire started outside of the congested district, developed into hot-blast form, and swept through and beyond the congested district, and burned out for lack of fuel.

8 On the other hand, the Baltimore fire began in the heart of the city and ramified more swiftly than the firemen could operate; then took the hot blast form and burned out for lack of fuel.

9 Such a hot blast has never been stopped by firemen while the wind held, but has, however, been checked and deflected upward by barriers consisting of two or more fire walls or their equivalent, with a free air space between, as per the known instances of fires out of control, which have been stopped by a mere alley if fully shuttered on each side.

10 It has also been possible to absorb the hot-blast attack of such fires by a very deep and fixed mass of spray in the form of sprinklered buildings. The Boston fire of 1893 was largely absorbed by an exceptionally good water supply in such form, and somewhat similar experiences have been had at Toronto in 1904, and other cases.

11 These successful experiences in checking hot-blast fires by deflecting the flame or by absorbing it in a mass of spray have been but little appreciated and instead of acting upon the lessons which they teach, our cities today have a collection of safeguards, part of which lend themselves to the debarment of conflagrations and part of which do not.

### SAFEGUARDS WHICH SINGLY CANNOT DEBAR CONFLAGRATIONS

12 The recognized and partly recognized safeguards against fire, no one of which alone can debar conflagrations, are twelve in number, viz., fire prevention; the fire limits; the water supply; the fire department; the high-pressure fire system; the uniform hose thread; the water curtain; the so-called fireproof building; the horizontally divided building; the protected window; the sprinklered building; and the piped building.

13 Dynamite, private hose, steam jets, carbonic gas systems, and fire walls separate from buildings are not listed because they are not recognized by fire chiefs for valid reasons.

14 *Fire Prevention.* The preponderance of disasters from trivial, unknown or unguessable causes appears to forbid hope of elimination of conflagrations through fire prevention. A half century of experience also shows that the skill and effort directed to prevent city fires from becoming disasters has been successful within 0.00003 of the total fires. This 0.00003 is what has hurt, and appears to be the only considerable task of correction remaining for the engineer.

15 *The Fire Limits.* This is an expression indicating a central territory, within which frame construction or shingle roofs are prohibited and better construction enforced, especially for large area buildings. The elimination of frame buildings and shingles is an essential part of any plan to debar conflagration.

16 *The Water Supply.* The water supply is indispensable and also adequate in most of our cities. Were this not so, Baltimore and other conflagrations would have gone to the extent of San Francisco, whose water mains were wrecked by earthquake.

17 *The Fire Department.* While no fire department has ever been able to put water on the front or rear of any hot-blast type of conflagration, nevertheless, the fire departments at time of conflagration have been of gigantic value in keeping the fire from spreading across the wind, in extinguishing brands thrown far ahead, etc. The fire department, therefore, is indispensable and this paper argues to increase its opportunity, but not its cost and size.

18 *The High-Pressure System.* Those cities which are partly equipped with costly high-pressure systems enjoy an advertisement which does not appear to be shared nor courted by the group of valley cities with reservoirs on high bluffs or hills. The latter afford a high-pressure service for the whole, not parts of such cities. Powerful high-pressure hydrant systems have also long existed in a goodly number of cities in the form of special inland hydrant lines operated by fire boats.

19 High-pressure develops long and large hose streams, but there is nothing about it to enable firemen to use such hose streams either in advance, or in the rear of, a hot-blast conflagration, nor is a high-pressure system especially flexible to check the ramification of fire as at Baltimore in 1904.

20 *The Uniform Hose Thread.* Since the fire department is indispensable, ability to double up departments is obviously wise and Mr. Griswold's long labors and the report of the sub-committee on fire protection on this subject are exactly in point.

21 *The Water Curtain.* It is fair to say that a water curtain

has successfully held off some heavy fires, but never a conflagration. The fatal weakness of the water curtain is that it is blown away or scattered by the brisk breeze which necessarily accompanies a hot-blast conflagration. The depth of the spray is also far too shallow.

22 The water curtain, as shown by fire record results, is a valuable safeguard for moderate exposure fires, or even for fairly severe exposure when favored by quiet air, but fails in severe tests.

23 *The Fireproof Building.* The typical, so-called "fireproof" building, having merely incombustible floors, roof and walls, cannot debar a conflagration (because of its unprotected windows and large volume of contents) any more than can an ordinary brick building with a good roof.

24 A hot-blast conflagration moves laterally, and a "fireproof" building in its path, as evidenced at Baltimore, is merely a crate which holds up the fuel contents in position for free burning and augments the general hot blast.

25 It is contents, not buildings, which make the bulk of property loss; contents, not buildings, which are hazardous; and contents cause buildings to burn in most cases.

26 Repeated experience shows that no building can withstand the heat due to burning any large quantity of contents or even very moderate contents if in a large rotunda, half floor office or like large area. The writer submits to all who are authorities on heat the feasibility of constructing a high structure of large retorts, each capable of restraining 10,000 to 50,000 lb. of ignited fuel, and also filling with customary taste and beauty the needs of utility and health for habitation.

27 The fireproof building has unequalled habitability and utility and also lends itself admirably to conversion into a piped building, and even more excellently lends itself to conversion into a building having protected windows.

28 *The Horizontally Divided Building* is designed to hold a fire from rising through the floors. Repeated experience in tall buildings has proven that excessively large and dangerous fires develop quickly if the floors are not fire tight.

29 Such horizontal division is obviously of no value against a hot-blast fire moving laterally from without, and it also does not prevent an internal fire from jumping from floor to floor on the outside of the building.

30 Horizontal fire-tight construction is very rarely found when

fire-tested, its worst drawback being that in the opinion of the public it injures the habitability and utility of the building, hence all sorts of concessions are made.

31 *The Protected Window.* Protected windows can debar city conflagrations, but do not because there are too few of them, except in a few special and minor localities.

32 *The Piped Building.* The building piped with fusible outlets, whether with automatic water supply or with water promptly applied, can but does not debar city conflagrations for the same reason given respecting the protected window.

TRUE HOT-BLAST CONFLAGRATIONS CAN BE DEFLECTED OR ABSORBED

33 Hot-blast conflagrations have been successfully controlled both by deflection and absorption.

34 Taking up the deflection idea first, all will doubtless agree that if the four walls of every city building were solid brick with no doors or windows, a spread of fire would be impossible, even with no fire department. From this it follows that if all doors and windows were protected by wire glass, shutters or fire curtains, the walls of the buildings would have the same tendency. Any experienced fire chief will testify to the enormous fire-stopping effect of an alley shuttered on both sides, the heat and hot blast being deflected upward.

35 Experience shows, however, that when a hot blast reaches a building prepared to act as a deflector by suitable protection for its doors and windows, the first result is partial failure. This is owing to the fact that the heat will radiate through the wired glass, or leak through the shutters, and ignite the contents.

36 Nevertheless, there will be a retardant effect and it is obvious that if other buildings located to the right or left of the center of the hot blast have window openings, similarly stopped, they must suffer less and the elsewise lateral ramification of the fire decrease.

37 The hot blast is thus largely deflected upward, partly checked and less able to cross the next street or alley, assuming protected windows throughout.

38 Just how many deflector walls and air spaces could be jumped or burned through by a conflagration of given severity is a matter of judgment based on observation, precisely as the extinguishing power of a hose stream is a matter of judgment from experience, not reducible to exact figures.

39 The writer submits that if all the alley windows were pro-

tected and also all the street windows on the second floors and above in the solid three and four-story parts of a town, a conflagration from without could not then bore a hole or a bay into such a district deeper than through four deflector walls and across three air spaces, which would mean two blocks and three streets, of which one might be an alley.

40 Not that the fire would be put out, nor that tongues and fire brands would not have to be taken care of; but that the hot blast would be deflected upward so the firemen could take a front stand and the general advance and ramification of the fire subside to a state of normal fire department control.

### DEBARMENT OF CONFLAGRATIONS BY ABSORPTION

41 There remains but one other known means to regain control of a conflagration, that of absorbing the hot blast by means of the piped building. Experience has demonstrated that a hot blast can be absorbed by a spray if the spray be very deep and fairly housed from the wind as is true of the cage of spray represented by a sprinkler installation in full action in a building whose windows have burned out.

42 The most notable demonstration of this was the Brown-Durell sprinklered building at Boston in 1893. Inasmuch as this building became a single large cage of spray which absorbed the main body of a down-town fire that was wholly beyond control, it is certain that a row of such cages of spray, if placed two or more deep, would always accomplish the same thing, and do so without the aid of protected windows.

43 The writer submits that if a city throughout all of its three and four-story and higher parts were composed exclusively of suitably piped buildings, and special water supply provided, a conflagration from a district without could not burn across a street, through a block deep of spray, and across the next street.

44 The fire would not be put out and fire brands would have to be taken care of; but there would be no ramification of fire in the sprinklered territory and there would be a full restoration of normal fire department control.

### VALUES, COSTS, GAINS

45 The Boston big fires proved out burnable property values at a rate of over $500,000,000 per square mile, and it is well known that

today there are several city centers which have grown to a far higher rate of concentration of value.

46 It seems fair to assume $250,000,000 per square mile as an average burnable value over the central districts of our twenty leading cities.

47 For such a square mile, standard automatic sprinklers (including masonry) would cost about four per cent of the burnable values, or $10,000,000 with fixed charges of about 16 per cent per year.

48 Empty sprinklers, or, protected windows, would each cost

ABLE 1 FIXED CHARGES, FIRE COST SAVING AND NET GAIN ON A BASIS OF $250,000,000 PER SQUARE MILE

| On an Annual Basis | For Standard Automatic Sprinklers with Double Water Supply | For Empty Sprinklers and Piping to be Supplied by the Fire Department | For Shutters, Fire Curtains, or Wired Glass Applied as per Preceding Test |
|---|---|---|---|
| Investment per sq. mi. | $10,000,000 | $5,000,000 | $5,000,000 |
| Fixed charges as given above | 1,600,000 | 450,000 | 450,000 |
| Saved by eliminating risk of conflagration at 33 ct. per $100 | 825,000 | 825,000 | 825,000 |
| Saved by eliminating common exposure fires at 7 ct. per $100 | 175,000 | 175,000 | 175,000 |
| Saved by reducing fire cost within buildings in which fire originated | 2,000,000 | 1,125,000 | nominal |
| Difference between charges and savings | 1,400,000 | 1,675,000 | 550,000 |
| Per cent earned by savings | 14 | 33.5 | 11 |

about half as much, or $5,000,000 per square mile for either, and each incur fixed charges (about the same as the buildings), or about 9 per cent.

49 The savings per $100 of burnable values per year, would be as given in Table 1.

50 In surveying any actual square mile it would develop that but one of the three, viz., protected windows, automatic pipes, or empty sprinkler pipes, would best suit any one building, and this would be likely to result in a detail plan calling for gross investment of about three per cent of the burnable values at a net gain of about 18 per cent.

51 But figures cannot include the grief, loss of work and trade following every large conflagration.

52 Our fire limits, fire departments and waterworks are today well developed, and protected windows or piped buildings throughout the costlier districts are all that is needed, to debar conflagrations.

RECAPITULATION

53 To recapitulate the advantages and disadvantages of the protected window and the two types of piped buildings:

54 The protected window delays the entry of severe fires and also prevents general ramification of fire through innumerable window openings.

55 Not that the protected window does this perfectly, because shutters may be out of order or not within reach to close if open, and because wired glass transmits heat by radiation very rapidly. Nevertheless, as aided by existing air spaces, alleys and streets, the protected window is a proven success.

56 The protected window is beginning to be required in building codes; it also is tangible to the public eye, something that can be seen as representing a fire stop or check; in wired glass form it has some working advantages, at least for skylights, and finds favor with architects on the better class of buildings; in the form of shutters, the fire-stop effect is better than for wired glass, but this is largely offset by the fact that shutters do not get the care which comes to a window which is in more or less constant use.

57 The advantage of the piped building with automatic double-source water supply, the well known sprinklered building, is first of all the protection to life. Apparently this specific form of fire protection is the only one which to any dependable degree conserves life. An experience with say 10,000 buildings over a period of about 15 years gives rise to the statement that no life has ever been lost in a building so equipped, either by fire or smoke, and to the best of the writer's knowledge this is literally true.

58 The operation of an automatic sprinkler system develops a powerful drenching spray not only on the fire but around it, and compels escaping smoke to pass through a dense spray which takes up the acrid quality and heavier carbon contents of the smoke, and thus has much to do with the protection to life.

59 While mathematical safety against loss of life is probably impossible, it is within the truth to say that where people are in masses, or are asleep, safety cannot exist if the fire hazards are not under the automatic sprinkler.

60 A second advantage of the automatic sprinkler system, and the one most in point under the title of this paper, is that it has been found in practice that, given brick buildings, well secured pipes, and reasonable water supply, a fire even when of conflagration magnitude cannot burn completely through such spray further than the depth of one, or say two, buildings.

61 A third advantage is that the fires are put out so quickly and with such economy of water by reason of its accurate application, with so little smoke and so great a reduction of the harmful quality of the smoke that the aggregate fire, water, and smoke damage to goods is far less than for any other form of protection.

62 Referring to the plain piped building, or building equipped with fusible sprinklers on empty piping with exterior hose coupling for fire department use and relying solely on the fire department for water supply, the main advantage is that the first cost and low fixed charges make it applicable to the medium value buildings.

63 Another advantage is that of safety to life, compared with that of buildings not piped at all, because in practice the empty sprinklers can be operated nearly as quickly, and necessarily to the same effect as automatic water-supplied sprinklers.

64 Still another advantage is that the technique and upkeep essential to efficiency are far less than with the full standard automatic sprinkler equipment.

65 The main disadvantage of the protected window is that it protects only as between neighboring buildings and this saving averages too small an amount to cover its fixed charges, on a basis of every-day fires.

66 A disadvantage of the standard automatic sprinkler system is that it is a special engineering product, technical to a high degree, yet depending on this quality for its efficiency, an efficiency imparted by a few skilled contractors and experts. The system therefore is open to criticism by all who rail at any control of skill or service.

67 Another objection to this form of piped building is that its water supplies are often direct from city mains through the influence of large property owners, thereby saving them the expense of private water supply. In Manhattan Island and Chicago the city water is of too low a pressure to be so used, but in other cities there are too few sprinklered buildings to check a conflagration and just about enough of them to jeopardize complete crippling of waterworks and fire department at such a time by reason of these buildings being

wrecked and bleeding the general water supply throughout the breaking of large pipes.

68 Hence a wise requirement for a piped district, would be to provide a special border pipe line into which water would be pumped or admitted under control.

69 Still another disadvantage of standard automatic sprinkler equipment is in first cost and in fixed charges. The investment and fixed charges do not have any fairly constant relation to the value of building plus contents, and at city labor costs are usually excessive, except for large and costly buildings.

70 A disadvantage of the empty pipe sprinkler system is that this mode of protection has as yet but few applications; no extended study has been given the art of cheap extinguishment of fire in medium value property. Another disadvantage is that fire department practice is at variance; some chiefs favor and ask for such equipment, and others evade or object.

71 It does not seem to be generally realized that a building in a central district does not burn badly before the department arrives. Were this not so, modern fire departments would not, as the records show year after year, hold the fires within moderate loss, except 0.003 to 0.005 of the total. The fire department does arrive while the fires are yet incipient, though perhaps inaccessible. It seems to be accepted as a matter of course that a costly proportion of buildings shall burn and soak, subsequent to the arrival of the department, for the sole reason that the department cannot quickly put ample water where, and only where, it is needed. Yet to do the latter is all that the standard automatic sprinkler equipment does or professes to do, and water can be supplied to empty pipes nearly as quickly by firemen as by a private tank.

72 Even in a case of purposely delayed alarm and sprinklers shut off (incendiary), the writer has seen work done in this manner by only one steamer with wonderful success, extinguishing a four-story fire which otherwise would have required many hose streams, and this after there was no time to set up ladders and place hose.

73 The fire cost of empty sprinkler equipment would admitttedly be greater than for automatic sprinkler equipment because while there would be no failures through pipes frozen or valves shut off, the fire department would not put water on the fires at quite as early a stage of incipiency.

74 A willing fire department, however, would put water on the

fire through such pipes while a fire was yet incipient, because our fire department records show that the department arrives and the vast majority of city fires are put out while incipient.

75 To pay for the greater fire and water damage in practice with empty pipes as compared with automatic sprinkler equipment, the figures given in Table 1 allow for the typical square mile, $825,000 per year, to say nothing of $1,160,000 reduction of annual fixed charges.

76 However, the standard automatic sprinkler system has been fully demonstrated for over 20 years, yet it is but just coming into its own, and the empty sprinkler system must in turn wait for recognition and extend in application by degrees.

77 The practical difficulty, therefore, of debarring disastrous conflagrations in our cities seems not in lack of means, nor in lack of knowledge based on experience, so much as in the lack of agreement on a plan, and the difficulty of apportionment of expense.

No. 1393 *b*

# BALTIMORE HIGH-PRESSURE FIRE SERVICE

By James B. Scott, Baltimore, Md.

Member of the Society

*The Conflagration of 1904.* The conflagration of 1904 was due to the simultaneous occurrence of an incipient warehouse fire gaining headway unobserved on a Sunday morning; a high wind; inferior building construction and inadequate fire-fighting equipment.

2 During the first eight hours of the fire the wind was blowing from the southwest, after which it shifted 90 deg. to the northwest, causing the fire to advance with its broadside of 1500 ft. for a front. Although supplemented by engines from other cities, after the fire had got beyond control the operations of the department might be described as a skilful retreat, an engine and a truck being lost under falling walls because the retreat had not been sufficiently rapid. Dynamite was freely used by skilful operators, but was practically ineffective. The fire raged for 30 hours, covering 150 acres, causing a loss of $100,000,000 and finally burned itself out when the wind changed again to the north and drove the flames toward the open harbor.

3 There was no scarcity of the water supply. The topography of Baltimore shows elevations ranging from 6 to 460 ft. above mean low tide, and to prevent excessive pressures in the low lying sections or a deficiency in those higher up, the supply is divided into five separate services. The "low" and "middle" services are fed entirely by gravity, the three higher services being supplied by pumps and high storage reservoirs. Suitable by-passes are provided so that in an emergency any service can be supplied from the next higher. At the time of the conflagration there was available a total reservoir capacity of over 1,750,000,000 gal., in addition to pumps of 63,000,000 gal. capacity. At that time the consumption for domestic and industrial purposes was about 60,000,000 gal. daily, and the draft from the reservoirs for fighting the conflagration was approximately the same amount, or a trifle over 3 per cent of the available reservoir

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

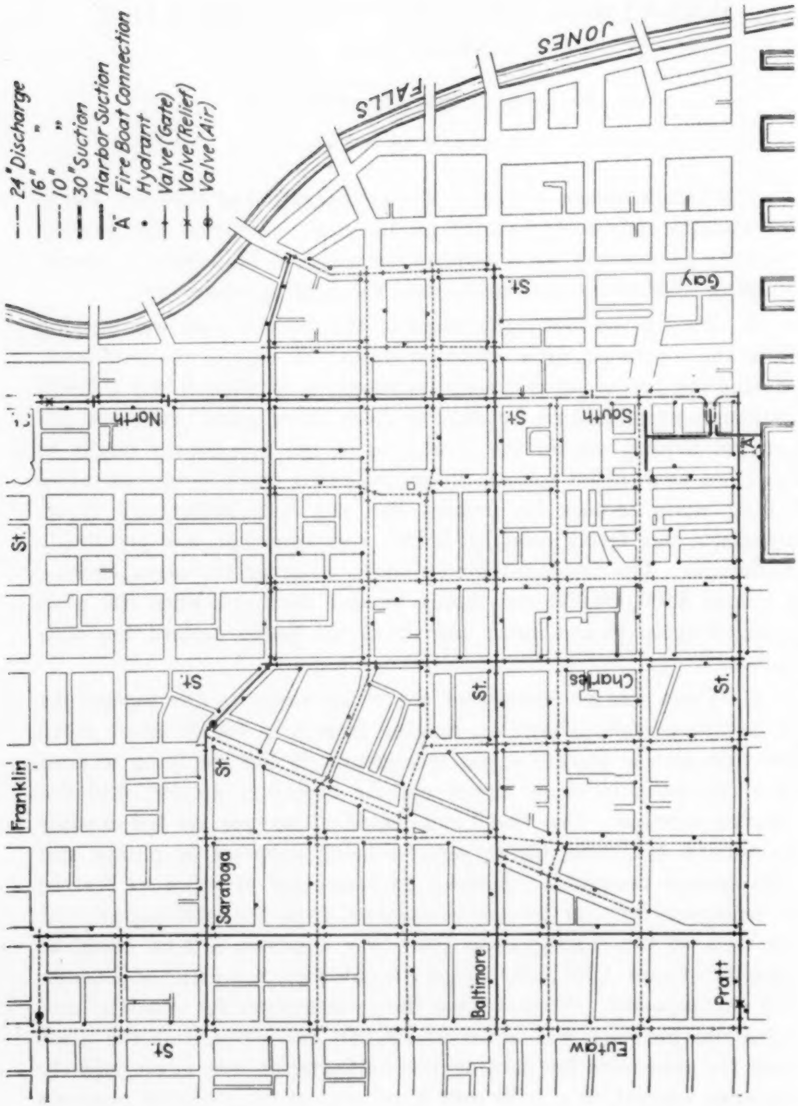



FIG. 1 MAP OF THE HIGH-PRESSURE DISTRICT

capacity. In addition to the reservoir draft, about two hours after the fire started the two 17,500,000-gal. pumps in one of the high-service pumping stations were by-passed into the "middle" or upper gravity service. The fire originated in the middle service district and shortly afterward extended into the low service district. As soon as this occurred the by-pass between the two was opened, and the pressure was maintained at 80 lb. during the remainder of the fire.

4 The operations against the conflagration demonstrated that, for effective fighting of a dangerous fire, large volumes of water must be delivered on the fire with the least possible delay, and at suitable pressures. To meet these requirements a separate high-pressure fire service was designed and installed, covering the greater portion of the congested value districts of the city.

5 *Characteristics of the Congested Value Districts.* The corporate limits of Baltimore embrace an area of 31½ sq. mi., with a density of population of 17,729 per sq. mi., a density greater than any other of the larger American cities. The congested value districts cover approximately 300 acres, the area at present covered by the high-pressure mains is about 175 acres, or assuming that the territory for 200 ft. outside the mains is protected, the high-pressure district may be considered to be 245 acres.

6 The elevations of the congested value districts vary from 6 ft. to 100 ft. above mean low tide, about one half of the area being below elevation 50. The pump centers at the high-pressure station are at about elevation 12.5.

7 The character of the building construction in the congested value districts varies from first-class modern fireproof structures to converted residences, except in the rebuilt burned district, where a better type of construction exists. In the districts in question all electric wires have been placed underground in conduits owned by the city, with the exception of the street railway trolleys. Before rebuilding the burned district, several narrow and congested streets were widened and plazas established around important public buildings, so that the conflagration hazard in that section has been considerably decreased.

## DESIGN OF THE HIGH-PRESSURE SYSTEM

8 *The Time Element.* High-speed automobile hose wagons are provided, housed at convenient locations, in order that the minimum

time possible may be consumed in reaching the scene of a fire after the alarm is received. Each wagon is equipped with a four-cylinder four-cycle motor of sufficient power to develop a speed of 30 mi. an hr. through the streets of the city, with a load of 5000 lb. Each body is 50 in. by 34 in. by 12 ft. inside, and carries 2000 ft. of 3-in. hose. Each body is equipped with one 2000-gal. Morse Invincible monitor nozzle and two 1100-gal. monitors.

9 *The Pressure Element.* The pressure required for effective work varies widely according as conditions demand the flooding out of a basement fire or fighting on the top floor of a modern skyscraper. At times both extremes may be required simultaneously.

10 To meet efficiently these conditions requires the maximum pressure to be available at each hydrant, with means for the separate control of the pressure on each hose line. The specifications called for a combination operating valve and regulator capable of adjustment from shut-off to 50, 75, 100, 125, or 150 lb., or if desired, to the full line pressure of 300 lb. The pressures were to be plainly marked on the valve by notches to be used by the operator as a guide for setting the handle. Regulators were to hold the pressures steadily within 10 lb. of the set amount, whether the play pipe were open or closed. The regulator was also to be provided with a lock whereby the handle could be prevented from passing the 150 lb. notch, but after unlocking could be moved to the full line position. When wide open under the maximum pressure of 300 lb. the valve was not to show a loss of head in excess of 15 lb. The hydrant head was to contain four horizontal outlets and one vertical outlet, each horizontal outlet to be provided with a regulator, and the complete head including four regulators was not to weigh over 110 lb. The head and regulators were to be designed for 300 lb. working pressure and were to be tested with a static pressure of 600 lb.

11 While the above requirements may seem simple enough, only one of the valves submitted by different makers met the requirements in all essential points, a regulator designed especially for the purpose by the Ross Valve Manufacturing Company of Troy, N. Y. (Figs. 2 and 3).

12 The main regulating valve is inserted in an opening just over the hose connection, and is inclined outward at an angle of 20 deg. The opening is closed with a plate carrying a pilot valve and a guide for the main valve. The pilot valve with its diaphragm is covered with a spring chamber, the whole being held in place by cap screws.

The main valve is balanced and is provided with a flat seat and leather face. The upper part of the main valve acts as an operating piston, being provided with a cup leather packing. The pilot valve is balanced against the delivery pressure by the regulator springs, which are made double to secure a wide range of pressure in a short length. The top of the spring chamber is revolved by the operating handle attached to it, and being provided with a coarse square thread screw, less than one revolution is sufficient to give the full range of pressure on the springs from full open to closed. The pilot valve is held positively in the two extreme positions independently of the springs by stops at the top and bottom of the stem. The full hydrant pressure is admitted to the operating chamber of the main valve through a small tube projecting below the seat of the valve. This tube is extended in order to keep the entrance clear of the varying velocity near the valve seat, which would tend to vary the flow of water to the operating chamber. When the pilot valve is open, water wastes from the operating chamber, the pressure is lowered and the main valve is opened by the unbalanced pressure below. When the pilot valve is closed the full hydrant pressure is maintained in the operating chamber, and as the area of the operating piston is somewhat larger than the area of the main valve, the pressure is unbalanced in the opposite direction and the main valve is closed. Intermediate positions of the pilot valve are followed by corresponding movements of the main valve, maintaining the delivery pressure within a few pounds of the amount indicated by the notch at which the operating handle is set. The main valve when fully opened presents an unobstructed waterway. The entire mechanism is simple in design, easy to operate and has proved entirely satisfactory in service.

13 *Quantity of Water Available.* The National Board of Fire Underwriters after a careful study of the situation recommended that a total delivery of 15,000 gal. per min. should be available within any area not exceeding 100,000 sq. ft., at a pressure of not less than 200 lb. at the hydrant. As installed, each hydrant has four 2½-in. horizontal and one vertical outlet for mounting a monitor nozzle. At present the 3-in. hose is connected through reducer couplings, but in the future all hydrants will be equipped with 3-in. outlets and connections to 2½-in. hose will be made through reducers.

14 The quantity of water which can be delivered through a hydrant is a function of the number, length and size of hose lines

attached, and the diameter and type of nozzles. Assuming four lines of 3-in. rubber-lined hose each 100 ft. in length, and 200 lb. at the hydrant, with $1\frac{3}{4}$-in. smooth nozzles, each hydrant would deliver 3800 gal. per min., or the requirement of the underwriters would be met by four hydrants. At least twice that number of hydrants are available for every unit of area mentioned. If 2-in. nozzles were used, other conditions remaining the same, the discharge would be 4400 gal. per min. at a pressure of 86 lb. at the base of the play pipe, as compared with 108 lb. in the former case.

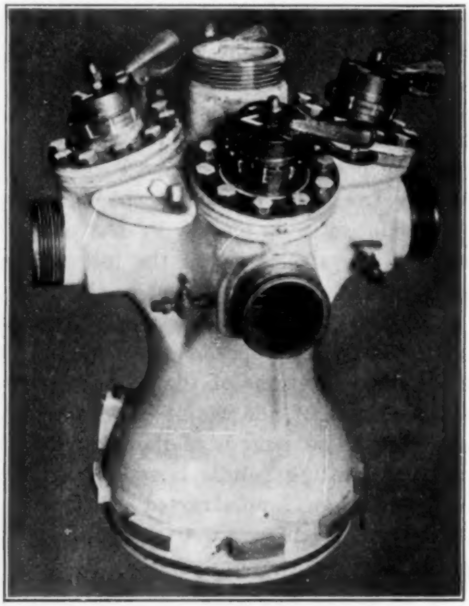


FIG. 2 PORTABLE HEAD

15 At present 226 hydrants have been installed, staggered on opposite sides of the streets, averaging about 170 ft. lineal spacing. In addition to the normal locations at street and alley intersections, the fire chief placed a number of hydrants at special positions, to meet local conditions of exposure, extra hazard or specially congested values.

16 *Design of Hydrant.* The type of hydrant to be used was given careful study. It was considered especially desirable in view of the experience gained in fighting the conflagration, to be able to

place a monitor or special flat nozzle directly on the top of the hydrant, to form a water curtain for the protection of exposed property on the opposite side of the street. Street intersections form a specially desirable location for such a purpose, but usually the corner of the footway is preëmpted for various other structures such as sewer inlets, lamp posts, trolley poles, police and fire-alarm boxes, etc. It was soon evident that if restricted to the ordinary post type of hydrant, it would be impossible to secure suitable locations, and the

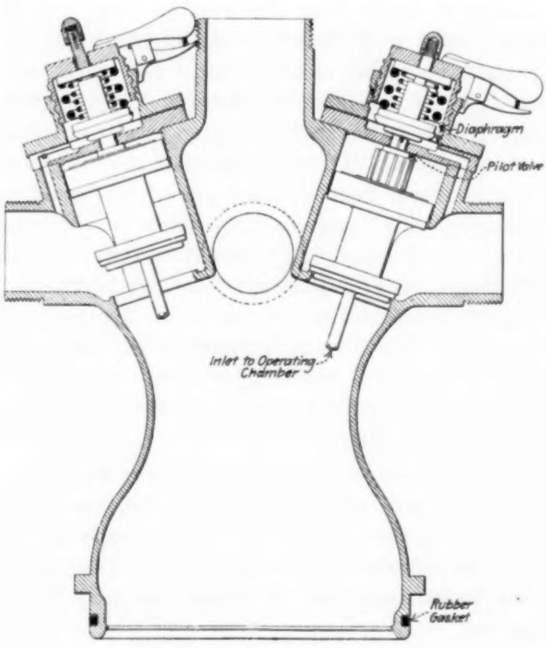



FIG. 3 SECTION OF PORTABLE HEAD

water-curtain feature would have to be abandoned. The flush hydrant with portable head seemed to meet all the conditions admirably. By the use of this type, hydrants could be located almost without restrictions, either in the driveway or any part of the footway. The portable head was also admirably adapted to the use of the regulating valves on the hydrant, as the entire operating mechanism, other than the main valve, could be kept in the firemen's quarters, instead of being exposed to frost or other injuries on the street.

17 The portable head as designed complete with four regulating valves, weighs 110 lb. In order to provide against any delay in attaching the head to the hydrant, a special "bayonet" joint was designed (Fig. 2). The head slips loosely into the barrel of the hydrant, and by a twist of 22½ deg., a series of interlocking lugs on the head and barrel engage each other, and the full section of these bronze lugs is in shear to resist the water pressure. The water joint is made by means of a square soft rubber packing ring placed in a square groove on the outside of the lower portion of the head. The groove is somewhat larger than the packing ring, and at very low pressures water leaks past the ring. At higher pressures, however, the water presses the rubber closely against the barrel, and the joint is absolutely tight at all pressures between 20 and 1000 lb. The action is entirely automatic, there being no screws nor glands of any kind to be manipulated. When the water is shut off, the ring contracts and the head can then be lifted out of the barrel without the slightest resistance. To illustrate the extreme simplicity of the device, and the ease of handling, a recent test by the engineers of the National Board of Fire Underwriters may be cited. The head and operating key were laid in the center of the street, 20 ft. away from the hydrant. Two firemen selected at random, picked up the head and key, ran to the hydrant, removed the two loose covers, placed the head in position and turned the water on, all in the space of 18 seconds by a stop watch.

18 A small cast-iron cover is laid over the top of the barrel to protect it from dirt and injury, and over this is placed a larger cast-iron cover flush with the pavement. As practically all the hydrants are located on the sidewalks the cover is made quite light, and if it should become frozen in, it can be broken instantly by a blow from the operating key.

19 The hydrant proper is designed for a clear waterway of 28 sq. in. through the main valve. The main valve closes with the pressure, and an auxiliary is provided, actuated by the main valve stem. This auxiliary valve opens in advance of and equalizes the pressure before the main valve starts to open. A drip valve is arranged so that as the main valve opens the drip is closed and vice versa, but both valves cannot be open at the same time. Both the main and auxiliary valves have conical leather faces and bronze seats. The barrel of the hydrant is of soft grey iron, and all nuts and fittings, stuffing-boxes, etc., are of bronze. The main valve stem is Tobin bronze, and the

operating spindle is of forged steel. All pressure parts are designed for a working pressure of 300 lb. with a factor of safety of twelve. After erection a field test of 600 lb. was made.

HIGH-PRESSURE PIPE LINES

20 *General Plan.* The general plan of the system is a gridiron of

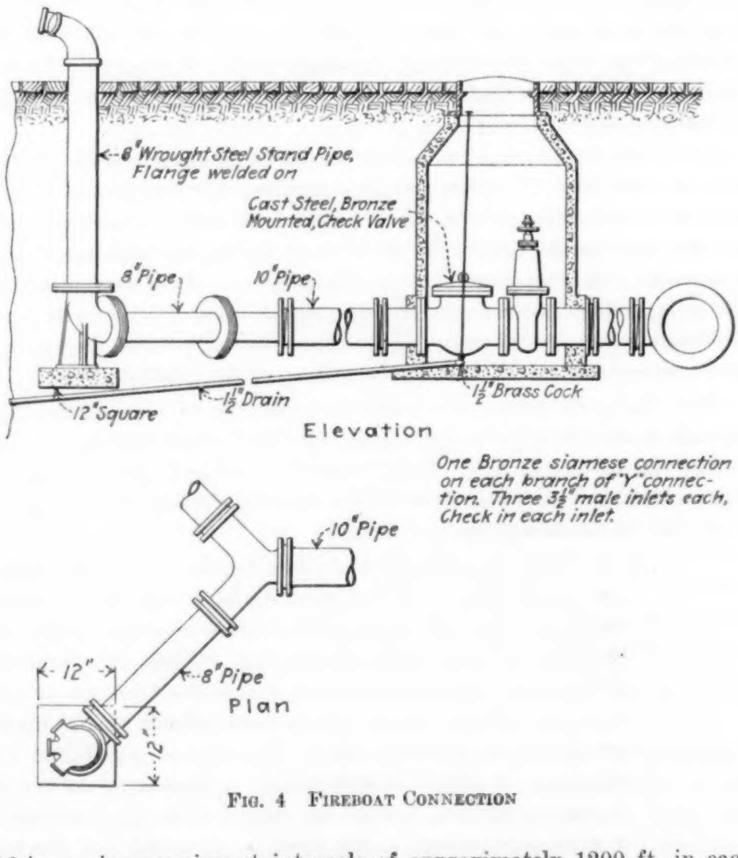



Fig. 4 Fireboat Connection

16-in. mains crossing at intervals of approximately 1200 ft. in each direction, with 10-in laterals in the intermediate streets spaced approximately 400 ft. apart. A 10-in. branch is provided at the harbor front for connection with the fireboats (Fig. 4). All hydrant branches are 8 in. There are no dead ends on mains or laterals in the entire system, with the exception of an extension on one street

which will be connected up at the next addition to the system. From the pumping station two 24-in. mains discharge into the intersections of the 16-in. mains. In the pumping station the 24-in. mains are looped around the rear of the pump foundations, thereby avoiding dead ends and by equalizing the stresses render unnecessary any heavy anchorages. Individual 14-in. pump discharges deliver directly into the 24-in. mains through welded necks. Three 24-in. valves and an 18-in. cross connection in the station, serve to sectionalize the large main, so that any portion may be cut out without putting more than one pump out of commission. A large rolled steel air chamber 30 in. in diameter by 20 ft. high is placed on each 24-in. discharge main.

21 *Cast-Iron versus Steel Pipe.* For the service in view it was believed that cast-iron pipe, usually employed for city water-mains, was not a scientific application of material for the stresses involved. On the other hand, commercial rolled steel lap-welded pipe meets the structural conditions much more satisfactorily. With this material the pipe system becomes an engineering structure, all tensile and bending stresses are taken up by the pipes themselves and the system would be entirely safe if laid on the surface of the ground.

22 The principal objection urged against the use of steel for this purpose is the liability to corrosion, especially when complicated by electrolysis in city streets. After a careful study of the problem, it was decided that the advantages of this material outweighed the objections, for the following reasons:

*a* If the entire pipe system had to be renewed every ten years, the steel would still be more reliable and hence more desirable than cast iron, as the latter is liable to fail by breakage without previous warning. When deterioration of the steel pipe does occur, it makes itself known by increase of leakage, which can be detected and repairs made gradually, as occasion offers. By way of a reductio ad absurdum it might be argued that if absence of corrosion be the controlling feature in the selection of a material, a glass pipe would be the ideal, as it would last forever if not broken by shock or bending stresses.

*b* The study of corrosion of steel has reached a point where it is possible to say that corrosion is not inevitable, but is due to more or less direct violation of certain well defined principles. If steel can be protected from the

simultaneous action of moisture, air and acids, the causes of rusting or corrosion are to a great degree removed, regardless of the particular chemical theory held by the investigator. Protecting the steel from the action of these three agents is mainly a mechanical proceeding. A permanent, impervious, elastic coating, which will adhere closely to the metal is the requirement. High-grade asphalt applied when both the steel and the coating are hot and clean, seems to meet these conditions satisfactorily. The specifications for the chemical composition of the asphalt used were very exacting. Among other conditions it was required that a cubic centimeter of the material should show no action when exposed for one year in any or all of the following solutions: 25 per cent hydrochloric acid, 25 per cent sulphuric acid, 25 per cent potassium cyanide, 25 per cent caustic soda, saturated solution of ammonia. The pipes were thoroughly cleaned and heated to a temperature of 300 deg. fahr. and while hot were dipped vertically in the bath of asphalt, which was maintained at a temperature of from 350 to 400 deg. fahr. The pipe was held in the bath for a sufficient length of time and was then drawn out slowly, at the rate of 5 to 10 ft. per min., so that a coating of 1/32 in. thickness was evenly distributed over the entire surface of the pipe. Any damage to the coating during shipment or erection was repaired by the application of the same material dissolved in a suitable solvent, and applied in several coats at intervals, until a satisfactory thickness was obtained. All bolts and nuts were dipped in the same solution before being inserted in the flanges.

*c* Electrolysis is due to an electrical difference of potential between the metal and the earth in contact with it, of sufficient magnitude to cause a current flow from the pipe to the earth. Since the conflagration of 1904 the City of Baltimore has required all electrical wires to be placed in the municipal conduits, and to protect the cables from electrolytic action the city's electrical commission made a careful study of the local situation. As a result the street railway was compelled to rebond a

large part of its tracks, copper cables being carried around all special work. In addition a supplementary copper return covering the entire district was installed, consisting of three bare copper cables having an aggregate cross-section of 6,000,000 circular mils. An entirely separate copper return was also installed as a protection for the cable sheaths, nothing but lead cables being bonded to this. The cross-section of this latter varied from 1,000,000 to 6,000,000 circular mils. As a result of this large amount of copper in the return circuit, the difference of potential between the various underground structures has been reduced to a nominal figure.

The special joint designed for the Baltimore steel pipe line has a resistance equal to 6 in. of the pipe on the 10-in. size, and equal to 9 in. on the 16-in. pipe. The resistance of a bell and spigot lead joint is often equal to 4 or 5 ft. of the pipe.

*d* Finally, the current flow from the pipe to the earth may be made very small if a high resistance covering be placed around the entire pipe system. In the present instance the asphalt coating furnishes the necessary high resistance envelope.

23 While it is still too early to offer definite evidence regarding the life of the pipe in question, pipes which have been in the ground for two years have recently been exposed by excavation for other work, and have shown absolutely no signs of deterioration, the coating being in perfect condition.

24 *Pipe.* The specifications called for lap-welded pipe made of soft open hearth steel having the following qualities:

| | Per Cent |
|---|---|
| Carbon, not exceeding | 0.10 |
| Phosphorous, not exceeding | 0.04 |
| Sulphur, not exceeding | 0.05 |
| Manganese between | 0.35–0.45 |

Ultimate tensile strength between 50,000 and 55,000 lb. per sq. in.; elastic limit at least ½ ultimate; elongation not less than 20 per cent in 8 in.; cold and quench bend 180 deg. flat.

25 The weld in the lap was to be perfect and capable of standing the strains incident to the manufacture of bends and forming of joints, without distress or rupture.

26 The thickness of pipe, in inches, was as follows:

| | |
|---|---|
| 24 in. Outside Diameter | ½ |
| 16 in. Outside Diameter | $\frac{7}{16}$ |
| 10 in. Inside Diameter | $\frac{7}{16}$ |
| 8 in. Inside Diameter | $\frac{7}{16}$ |

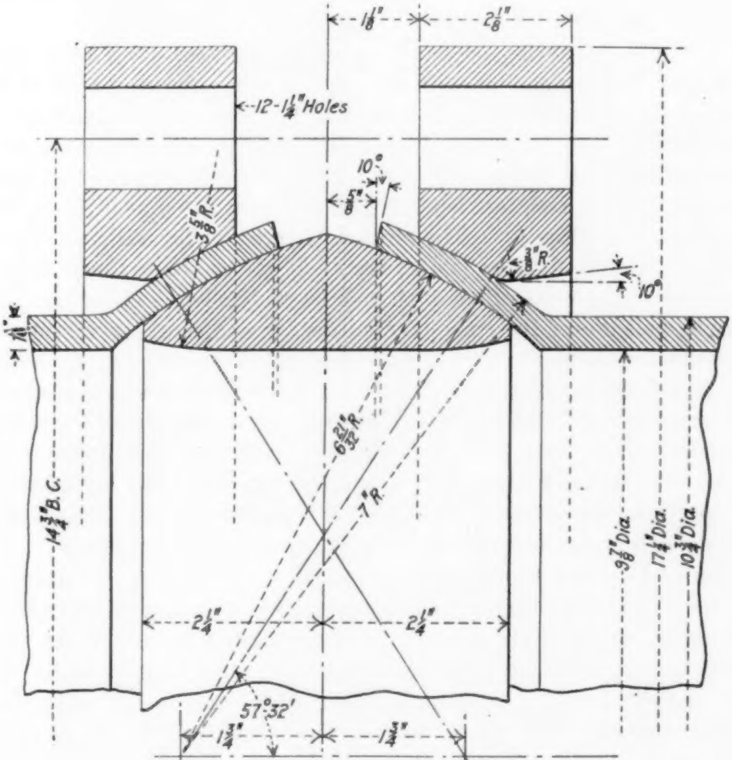


FIG. 5 DETAIL OF 10-IN. PIPE JOINT

27 Bends were generally of a standard radius of 5 diameters of pipe. Length of pipe, in feet, laid was:

| | |
|---|---|
| 24 in | 1,275 |
| 16 in | 17,052 |
| 10 in | 28,229 |
| 8 in | 7,137 |
| Total | 53,693 = 10.2 miles |

28 *Joints.* For the present service it was desired to avoid the use of rubber gaskets, which would tend to increase the electrical resistance of the joint. Copper gaskets or other metals dissimilar to the metal of the pipes, in the presence of moisture and the acids or salts of the earth, would form a voltaic cell, and tend to increase the corrosion due to electrical action. A joint was therefore designed to avoid the necessity for a gasket or joint cement of any kind.

29 The Baltimore design is a form of universal joint (Fig. 5). The end of the pipe is flanged out into a bell forming a zone of a sphere. A soft cast-iron ring is accurately turned in the shape of a torus, having the same curvature on its exterior surfaces as the interior of the bell on the pipe ends. Loose flanges are placed on the pipe back of the bell, and when bolted up, draw the pipe bells up on the torus ring (Fig. 6). The pressure secured by the wedging effect on the spherical surfaces is enormous. If the curve of the surfaces be too flat, the metal of the pipe may be cold rolled, and the flanges may be pulled over the ring. By suiting the degree of the curvature to the diameter of the pipe, a safe combination is secured, and with the proper thickness of loose flange, a joint is secured which is absolutely water tight until pressures are reached which exceed the elastic limit of the pipe or bolts. On the 10-in. size this pressure is about 2200 lb. per sq. in. In the field testing, during the installation of the pipe, the specified test pressure was 600 lb., but during the early stages of the work, this pressure was often largely exceeded.

30 The joints are designed for a deflection of 10 deg., or about 3 ft. 6 in. in a 20-ft. length. On the 10-in. size this amount of deflection is easily obtainable, but on the larger sizes a smaller amount was used, though sufficient to be of considerable value in city work. A line of 10-in. pipe was made up of seven 20-ft. lengths, and while the pressure was on, one end of the line was raised 5 ft. by means of a crane, as shown in Fig. 7. While suspended in the air, supported only at the two ends, 140 ft. apart, there was practically no leakage at any of the joints. It is evident that at least that amount of trench might be washed out without interfering with the operation of the pipe system in the least degree. In laying the pipe it was possible to deflect it to pass an obstruction and in one instance, after the completion of the work, owing to a change of grade in the sidewalk, a hydrant branch 50 ft. long was jacked up 5 in. at one end, simply by loosening the bolts at one joint. Upon tightening these bolts, the line tested free of leaks at 600 lb.

Fig. 6 View of Pipe Joint and Welded Neck

Fig. 7 Shop Test of Pipe Joint

31 The fact that the pipe was laid in 20-ft. lengths means that the number of joints was reduced 40 per cent in comparison with cast iron.

32 For deflections greater than could conveniently be made by the joints, pipe bends were used. A bending table was installed on the work by the contractor, and about 1500 bends were made, or 150 per mile of pipe laid. Bends of 16-in. pipe and smaller were made

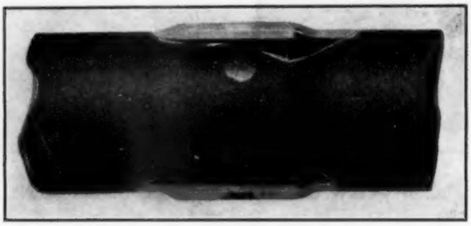


FIG. 8 DETAIL OF STRAIGHT LINE WELD

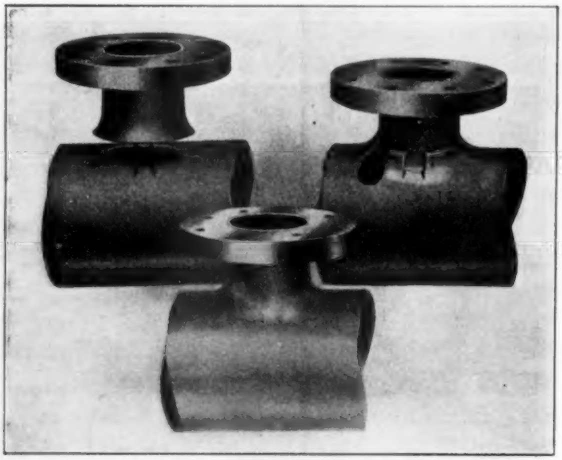


FIG. 9 DETAIL OF WELDED NECK

on the work, the 24-in. bends were made at the factory before shipment.

33 For make-up pieces at intersections and in blocks between valves or tees already installed, straight line welded joints were used. A special joint was devised for this service, made up as follows: The end of one pipe was accurately expanded sufficiently to

permit of its being shrunk over the end of the pipe to which it was to be joined. Holes were cut around the circumference of the outer pipe or bell, and after being heated it was shrunk on in place. The holes were then flowed up with metal by the oxy-acetylene blow-pipe, and the end of the bell was also welded to the enclosed pipe. With this type of joint the weld is in shear and not in tension, and it is entirely feasible to make bends in the pipe with the weld in the arc of the curve. Approximately 1500 of these straight line welds were made on the work, or about 150 per mile of pipe (Fig. 8).

FIG. 10 CAST STEEL SPECIALS

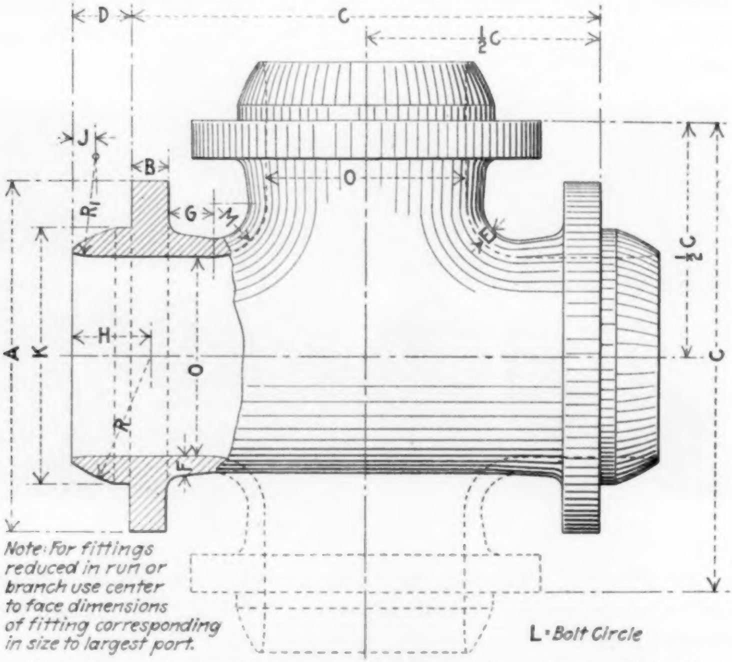
34 *Flanges.* The loose ring flanges were of medium open hearth steel, ultimate tensile strength 60,000 to 68,000 lb. All flanges were accurately machined and were drilled to template.

35 *Fittings.* The use of steel pipe made it possible to reduce the number of fittings to a minimum. The hydrant branches were made by welding necks to the mains and laterals. A special neck weld was used, made up as follows: A hole was cut in the pipe, smaller than the size of the neck, and radial cuts were made forming four narrow lugs which were left projecting into the hole. The wider alternate lugs were bent back to make an opening large enough to receive the neck piece. The smaller straight lugs formed a support for, and held the neck rigidly in place during the welding process. The whole joint was then flowed with metal by an oxy-acetylene blow-pipe, forming a joint as strong as the original pipes (Fig. 9).

36 The only specials required were the tees and crosses at the intersections of the gridiron, and a valve connection piece used to

give a straight face flange at the valves. By the use of the latter it was possible to use standard commercial valves, and the valves may be easily removed when necessary (Figs. 10, 11, 12). All fittings were made of low carbon open hearth cast steel.

37 *Valves* All valves are double-disk parallel seat gates. Bodies are low carbon open hearth steel of the same specifications as for

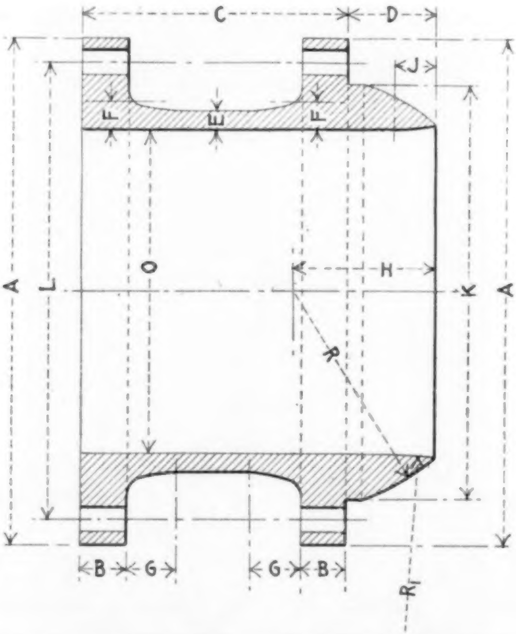



| SIZE | A | B | C | D | E | F | G | H | J | K | O | R | $R_1$ | L | M | BOLTS NO. | BOLTS DIA. | BOLTS HOLE DIA. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 10" | $17\frac{1}{4}$" | $2\frac{1}{8}$" | $23\frac{1}{2}$" | 3" | $\frac{13}{16}$" | $1\frac{1}{8}$" | $2\frac{1}{4}$" | 4" | $1\frac{1}{8}$" | $12\frac{27}{32}$" | 10" | $6\frac{21}{32}$" | $3\frac{5}{8}$" | $14\frac{3}{4}$" | $2\frac{5}{8}$" | 12 | $1\frac{1}{8}$" | $1\frac{1}{4}$" |
| 16" | $23\frac{1}{2}$" | $2\frac{1}{2}$" | 30" | $4\frac{3}{8}$" | $\frac{7}{8}$" | $1\frac{1}{4}$" | $2\frac{1}{2}$" | 6" | $1\frac{1}{2}$" | $20\frac{13}{32}$" | $15\frac{1}{8}$" | $9\frac{29}{32}$" | $6\frac{1}{4}$" | 21" | $2\frac{3}{4}$" | 16 | $1\frac{1}{8}$" | $1\frac{1}{4}$" |
| 24" | 33" | $2\frac{3}{4}$" | 41" | $3\frac{7}{8}$" | $1\frac{1}{16}$" | $1\frac{1}{2}$" | 3" | $8\frac{3}{8}$" | $2\frac{1}{4}$" | 27" | 23" | $14\frac{19}{32}$" | 7" | 30" | 3" | 24 | $1\frac{1}{4}$" | $1\frac{3}{8}$" |

FIG. 11 DETAIL OF TEES AND CROSSES

fittings. Bonnets, disks, gearing brackets, glands and packing boxes are of semi-steel. Stems are of Tobin bronze, not less than 53,000 lb. tensile strength. Seats, wedge mechanism and glands are of bronze. The 16-in. and 18-in. valves are geared. All street valves are provided with a stem nut, and a forged steel key is placed in each valve box. An interlocking arrangement is provided so that the key

can be removed from the nut only when the valve is wide open. This prevents the valves being left closed or opened only a few turns, except in an emergency or intentionally, as the valve box cover cannot be replaced while the key is on the nut. The 24-in. valves are in the pumping station and are hydraulically operated. All valves were subjected to a shop test of 800 lb. when open, and 600 lb. when closed.




| SIZE | A | B | C | D | E | F | G | H | J | K | R | $R_1$ | O | L | BOLTS NO. | BOLTS DIA. | BOLTS HOLE DIA. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 8" | $14\frac{3}{4}$" | $1\frac{7}{8}$" | $10\frac{1}{2}$" | $2\frac{15}{16}$" | $\frac{3}{4}$" | $1\frac{1}{8}$" | $2\frac{1}{4}$" | $3\frac{1}{4}$" | 1" | $10\frac{9}{16}$" | $5\frac{1}{4}$" | $3\frac{1}{2}$" | $7\frac{3}{4}$" | $12\frac{1}{4}$" | 8 | $1\frac{1}{8}$" | $1\frac{1}{4}$" |
| 10" | $17\frac{1}{4}$" | $2\frac{1}{8}$" | $11\frac{1}{2}$" | 3" | $\frac{3}{4}$" | $1\frac{1}{8}$" | $2\frac{1}{4}$" | 4" | $1\frac{1}{8}$" | $12\frac{27}{32}$" | $6\frac{21}{32}$" | $3\frac{5}{8}$" | $9\frac{7}{8}$" | $14\frac{3}{4}$" | 12 | $1\frac{1}{8}$" | $1\frac{1}{4}$" |
| 16" | $23\frac{1}{2}$" | $2\frac{1}{2}$" | 13" | $4\frac{3}{8}$" | $\frac{7}{8}$" | $1\frac{1}{4}$" | $2\frac{1}{2}$" | 6" | $1\frac{1}{2}$" | $20\frac{13}{32}$" | $9\frac{29}{32}$" | $6\frac{1}{4}$" | 16" | 21" | 16 | $1\frac{1}{8}$" | $1\frac{1}{4}$" |
| 24" | 33" | $2\frac{3}{4}$" | $14\frac{1}{2}$" | $3\frac{7}{8}$" | $1\frac{1}{16}$" | $1\frac{1}{2}$" | 3" | $8\frac{3}{8}$" | $2\frac{1}{4}$" | 27" | $14\frac{19}{32}$" | 7" | 23" | 30" | 24 | $1\frac{1}{4}$" | $1\frac{3}{8}$" |

Fig. 12 Detail of Valve Connecting Piece

38 *Valve Boxes.* The street valves are treated as pieces of mechanism, subject to derangement, and hence are placed in concrete boxes or manholes, 42 in. inside diameter, where they can be inspected and repaired without disturbing the street paving. The

floor was first laid of plain concrete, with suitable drainage connection; then the section enveloping the pipe or fitting was laid up with concrete blocks molded to the standard radius. The upper clear section was built of reinforced-concrete rings, laid up without mortar, as they are provided with interlocking projections on the top and bottom faces, which make the rings self-centering. The upper portion is made of conical shaped rings, forming the roof, and supporting the cast-iron cover frame. The blocks and rings were made up in quantities in metal molds and seasoned before being used. They

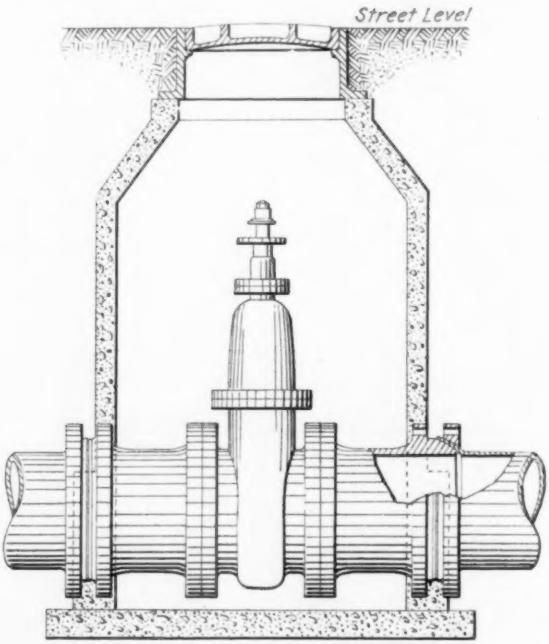



FIG. 13 VALVE BOX WITH VALVE IN PLACE

were quickly and easily assembled in the trench by unskilled labor. The holes were back-filled and the street repaved in a remarkably short time, an item of considerable importance in a business thoroughfare as there was no time required for setting of the concrete (see Figs. 13 and 14).

LOW-PRESSURE WATER SUPPLY

39 Two separate 40-in. mains from two different river systems, run direct to and from the normal supply for the high-pressure pump-

ing station. The normal pressure available with the high-pressure draft included is from 30 to 50 lb., but if by-passed from the middle or high-service mains, the pressure can be carried at 70 lb. A 30-in. cast-iron branch is taken to the station where it changes to a 30-in. rivetted steel pipe inside the building.

40 An auxiliary supply of brackish water from the harbor may be secured through a reinforced-concrete conduit 3 ft. sq. inside.

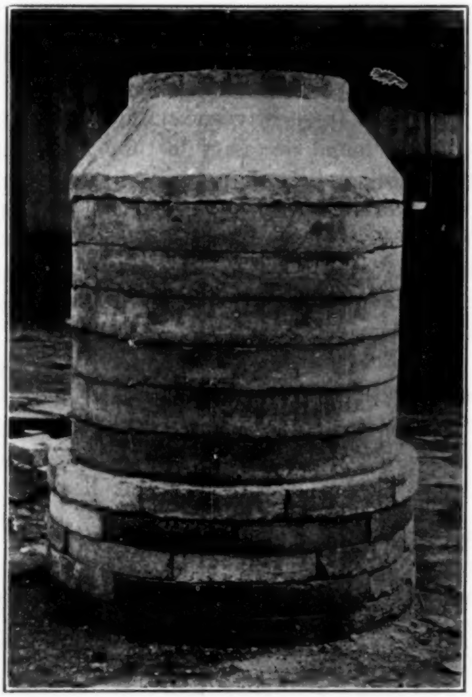


Fig. 14 Concrete Valve Box Made Up

The conduit is provided with a large screen chamber at the dock, and has fine and coarse screens sliding in inclined frames, and both in duplicate, so that one set of each can be in service while the other set is out for cleaning. The screen chamber is also provided with stop logs so that the chamber and conduit can be pumped dry for cleaning out sediment. At the station end the conduit terminates in a large suction chamber located beneath the basement floor and between the pump foundations. The dimensions of this chamber

are 8 ft. wide, 7 ft. deep and 70 ft. long. The gravity suction pipes enter the suction chamber through cast-iron thimbles built into the floor of the basement, as the basement floor is 8 or 9 ft. below flood tide level.

41 A by-pass from the high-pressure discharge main is led into the suction chamber for flushing it, and also forms a convenient discharge for the pumps during tests.

### INSTALLATION OF PIPE LINES

42 *Tunnels.* A considerable amount of tunneling was done, especially at street intersections, where double track crossings of the street railway were frequently encountered. On several of the busiest streets it was found necessary to tunnel for many blocks at a stretch. In a large number of instances it was thought desirable to tunnel for the hydrant branches, as the hydrants being staggered on opposite sides of the street, it was necessary on every alternate hydrant to cross the street, thus crossing also the railway tracks, pipes, sewers, electric conduits, etc. A segregation of these tunnels into groups is as follows:

| | Lineal Feet |
|---|---|
| Intersections | 5,122 |
| Straight line | 4,623 |
| Hydrant connections | 2,630 |
| Total tunnels | 12,375 |

43 *Leakage.* For field tests during the installation of the pipes, the contractor furnished a portable testing set consisting of a 3-h.p. 4-cycle gas engine, driving through a Morse chain a triplex plunger pump of a capacity of 5 gal. per min. at a pressure of 1000 lb. per sq. in. Each street section between valves was tested as the work was installed, and made absolutely tight, at the specified pressure of 600 lb: The section to be tested was filled from the ordinary fire hydrants, after which the test pump was connected through a 1-in. flexible steel hose. In the early stages of the construction, the workmen would frequently carry the pressure up to 800 lb. and sometime to 100 lb., to show interested spectators what the new joint would stand. Afterwards a relief valve set for about 610 lb. was placed on the pump, and the testing became almost automatic.

44 After the pipe lines had been completed, but before the system had been put into commission, a pressure of about 300 lb. was put on the pipe lines by the main pumps. There being no hydrants open,

the water was being wasted through a by-pass to reduce the pressure. A suggestion was made to close the pipe system off from the pumps to determine how long the pressure would remain. The main 24-in. valves were accordingly closed, while the pressure was at 125 lb. After a lapse of 16 hours the gage registered 95 lb., indicating that the leakage was practically nothing.

45 A duplex pump of 100 gal. capacity was installed to keep up the pressure and take care of the leakage on the system. Since the equipment has been put into regular service it has been necessary to keep the by-pass valves open continually on this pump in order to permit the plunger to move.

## CHOICE OF MOTIVE POWER

46 *General Considerations.* In the preliminary studies for the project, careful attention was given to the design of similar plants in other cities, notably New York and Philadelphia. In the former, as is well known, electric driven centrifugal pumps are used. In Philadelphia, gas engine driven, geared triplex plunger pumps were installed. In the following discussion it should be remembered that local conditions often exercise a controlling influence in the decision as to the best form of motive power for such a plant.

47 In Baltimore the fundamental principle governing the design of the entire project was considered to be reliability under the most adverse conditions liable to be encountered. When it is considered that the safety of hundreds of millions of dollars worth of property is dependent upon the accurate functioning of each element, it is evident that no considerations of economy, either in first cost or operating expense, should be allowed to enter, if simplicity and certainty of operation are thereby subordinated.

48 *Gas Engines.* The gas engine is subject to certain limitations, which in the case of fire service, place it at a distinct disadvantage.

- *a* The load factor of the service is extremely low, being less than 5 per cent per annum, so that the opportunity to profit by the high fuel economy is negligible.
- *b* Inability to carry more than a small percentage of overload.
- *c* In the hands of expert operators, a well designed modern gas engine is perhaps subject to no more accidents or delays than is a steam equipment, but certainly it is not subject to any *less* than steam. On the contrary, how-

ever, in the hands of operators of only average intelligence or experience, the number of apparently trivial causes which can result in serious delays or damage is surprisingly large in a gas engine plant. In the case of a municipal plant of this type it is useless to plan for ideal operating conditions, or to assume that only the highest grade operating force will be employed.

49 *Electric Motors.* In spite of its many known advantages, a large electrical distributing system is essentially in unstable equilibrium, and subject to complete interruption from very slight causes, either natural or malicious. Dependence for prompt renewal of the service after a shutdown must be placed in duplicate lines and equipment in the stations. But even with these a considerable period must elapse before large electrical machines can be started up and brought into synchronism. A delay of this character during the first critical minutes of a bad fire would be fatal to the usefulness of an important fire fighting system. The fact that a vital feature of the fire fighting system would be under the control of employes of an outside corporation and not subject to the discipline of the fire department, with the possibility of a conflict of authority at a crucial moment, are all matters that must be considered.

50 Like the gas engine, the electric transmission, especially the underground system, requires for its commercially profitable conditions a high load factor, but for a different reason. To justify the large investment needed, requires a uniform load as near as possible to its capacity, in order that the fixed charges shall not be out of proportion to the earnings. Owing to the extremely low load factor of a fire station, its load is a very "undesirable" one to a commercial electrical supply corporation, unless a "demand" charge is made sufficient to justify holding in reserve a definite proportion of the entire equipment from the coal pile to the electric cables. Electrical plants require the installation of sufficient equipment all along the line to supply the maximum annual peaks. As a fire is no respector of anybody's peaks, it follows that a fire station load demands its own separate power plant investment, whether the equipment is located in a commercial central station, or in an isolated plant for its own use.

51 A commercial plant is designed with a definite load factor in view, ranging usually from 30 to 50 per cent. For these conditions the most efficient equipment is justified, with all the refinements of

modern fuel and labor saving auxiliaries in a large plant. If a load factor of only 5 per cent had been imposed, however, a very different type of equipment would have been selected, at a very much smaller investment.

52 The various links in the chain of an electrical supply equipment (not considering a long distance transmission) would be as follows:

*a* Coal handling apparatus
*b* Boilers and auxiliaries (stokers, stacks, economizers, heaters, etc.)
*c* Steam turbines and auxiliaries
*d* High-tension generators
*e* High-tension switching apparatus
*f* High-tension cables

and at the receiving end

*g* High-tension switches
*h* High-tension motors
*i* Centrifugal pumps

53 If a long distance transmission were included, there would be added step-up transformers, aerial lines, a substation with step-down transformers and switching apparatus. If low-tension motors were used, additional transformers would be required.

54 An isolated steam pumping station, designed for the purpose, eliminates at once five of the above links, namely, items *d, e, f, g* and *h,* and concentrates the entire operation in one building, under the direct control of the fire fighters themselves.

55 *Finances of Steam and Electrical Operation.* Expressed in dollars and cents the argument becomes as follows:

Electric Pumps (New York Type)

Investment

| | | |
|---|---|---|
| 5 motor-driven pumps (rated capacity 3000 gal. per min.), switchboard, etc | $112,500 | |
| Building and pump foundations | 84,000 | |
| | | $196,500 |

Operation

| | |
|---|---|
| Maintaining pressure continually 8760 hr. less 100 hr. = 8660 hr. at 100 kw | 866,000 kw-hr. |
| Fire service, 100 hr. per annum 3150 kw. demand | 315,000 kw-hr. |
| | 1,181,000 kw-hr. |

| | | | |
|---|---|---|---|
| Service charge, maximum demand = | | 3150 kw. | |
| Central station investment, 3150 kw. at $75 | | $236,000 | |
| Underground cable (Baltimore conditions) | | 40,000 | |
| Cash requirements | | 276,000 | |
| Underwriting at 90 | | 31,000 | |
| Total investment | | | $307,000 |
| Fixed charges on $307,000 | | | |
| Interest | at 5 per cent | | |
| Depreciation | at 5 per cent | | |
| Profit | at 5 per cent | | |
| Total | 15 per cent | $46,000 | |
| Underground conduits, duct rental (Baltimore conditions) | | 1,300 | |
| Total service charge | | | $47,300 |
| Operating expenses | | | |
| Service charge | | $43,700 | |
| Meter charge, 1,181,000 kw-hr. at 1 cent | | 11,810 | |
| Salaries, station operating force | | 10,650 | |
| Supplies, lubrication and repairs | | 1,000 | |
| | | | $67,160 |
| Fixed charges on $196,500 | | | |
| Interest at 4 per cent | | $7,860 | |
| Depreciation at 5 per cent | | 9,825 | |
| | | | 17,685 |
| Total annual expense, electrical plant | | | $84,845 |

STEAM PUMPS

| | | |
|---|---|---|
| Investment | | |
| Four 4000-gal. pumps and auxiliaries | $86,000 | |
| Boilers and auxiliaries | 70,000 | |
| Piping, steam and auxiliary water | 30,000 | |
| | $186,000 | |
| Building and machinery foundations | 125,000 | |
| | | $311,000 |
| Operation | | |
| Coal consumption | | |
| Banking fires, 8760 hr.— 100 hr. = 8660 hr. = 360 days at 6 tons per day | 2160 tons | |
| Fire service, 100 hr. per annum at 5 tons coal per hour | 500 tons | |
| Total | 2660 tons | |

| | | |
|---|---|---|
| Operating expenses | | |
| Coal, 2660 tons at \$3.30 | \$8,778 | |
| Salaries, station operating force | 13,350 | |
| Supplies, lubrication and repairs | 2,000 | |
| | | \$24,128 |
| Fixed charges on \$311,000 | | |
| Interest at 4 per cent | \$12,440 | |
| Depreciation at 5 per cent | 15,550 | |
| | | 27,990 |
| | | \$52,118 |

| | | |
|---|---|---|
| SUMMARY | | |
| Total annual expense, electrical plant | \$84,845 | |
| Total annual expense, steam plant | 52,118 | |
| Total annual saving | | \$32,727 |

This saving capitalized at 9 per cent represents an investment by the city of \$363,630, considerably more than the first cost of the steam plant in the above comparison.

### STEAM PUMPING STATION

56 *Pumps.* The Baltimore plant is designed for four main units of 4000 gal. per min. rated capacity, at a piston speed of 300 ft. per min. and making 50 r.p.m. Three main engines have been installed and are in operation at present; the fourth unit will be added in the near future (Fig. 15).

57 In line with the policy of designing all parts of the system with a first requisite of simplicity and reliability, the main units each consist of a horizontal, twin, simple, non-condensing, crank and fly-wheel, plunger pumping engine. The water ends are attached directly to the engine frames, at opposite ends from the steam cylinders, the crankshaft being in the center (Fig. 16).

58 The steam cylinders are fitted with standard Corliss valve gears, having double eccentric long range cut-offs. The cut-offs of both cylinders are under the direct control of the speed and pressure regulators, and are also provided with hand control.

59 Each engine was liberally designed for a continuous working pressure of 300 lb. per sq. in. on the water ends, with a test pressure of 600 lb. static. Injection parts were designed for a variation of pressure from 70 lb. direct to a suction lift of 15 ft. of salt water. All steam parts were designed for maximum working pressure of 200 lb., but the normal working pressure is only 125 lb.

60 Each engine is fitted with speed and pressure governors. The speed governor is driven by a noiseless chain belt, and acts directly on the cut-off valves of both cylinders. The pressure governors are identical in principle with the regulating valves used on the hydrants, and were also furnished by the Ross Valve Manufacturing Company.

61 The net effective valve area between the openings of the valve seats on each suction and discharge deck is 308 sq. in. or 220 per cent of the cross sectional area of the plungers. The valves are $3\frac{1}{2}$ in. in diameter and are composed of rubber with brass backing plates; the seats are of bronze, screwed into the valve decks on a taper and faced off after being placed in the decks.

62 *Auxiliaries.* To take care of the leakage in the pipe system and maintain a pressure of 150 lb., as well as to provide for the first draft from the hydrants before the main pumps are in action, a 1000-gal. per min. pump was installed. This is a horizontal, duplex, direct-acting, compound, non-condensing, center-packed plunger pump, giving its rated capacity at a piston speed of 100 ft. per min. While the pump was designed for a normal working pressure of 150 lb. on the water end, and for 125 lb. on the steam end, all pressure parts were designed to withstand a continuous pressure of 300 lb. on the water end and 200 lb. on the steam end, so that the pump could be left in service under the maximum pressures of the main pumps without injury. Owing to the leakage on the system being so small, the pump is inconveniently large for the purpose intended, and it has been necessary to keep the delivery by-passed continuously in order to keep the plunger in motion.

63 In order to maintain the air in the delivery air chambers, there are provided two steam driven, crank and flywheel, two-stage air compressors, each having a capacity of 50 cu. ft. of free air per min. against a pressure of 450 lb. There is also provided a wrought steel storage tank 30 in. in diameter by 5 ft. high. There are two large air chambers on the 24-in. discharge mains, in addition to those on the pumps. The former are made of lap-welded rolled steel pipe, $\frac{1}{2}$ in. thick, 30 in. in diameter and 20 ft. high. The ends are bumped, riveted and welded to the pipe. Two sets of glass gages are attached through bosses welded on the sides.

64 For priming purposes, when lifting from the harbor, there is provided one vertical, steam driven, crank and flywheel single-stage, dry vacuum pump, 6 in. steam, 10 in. air, and 6 in. stroke. This pump is connected to the harbor suctions through a large separator

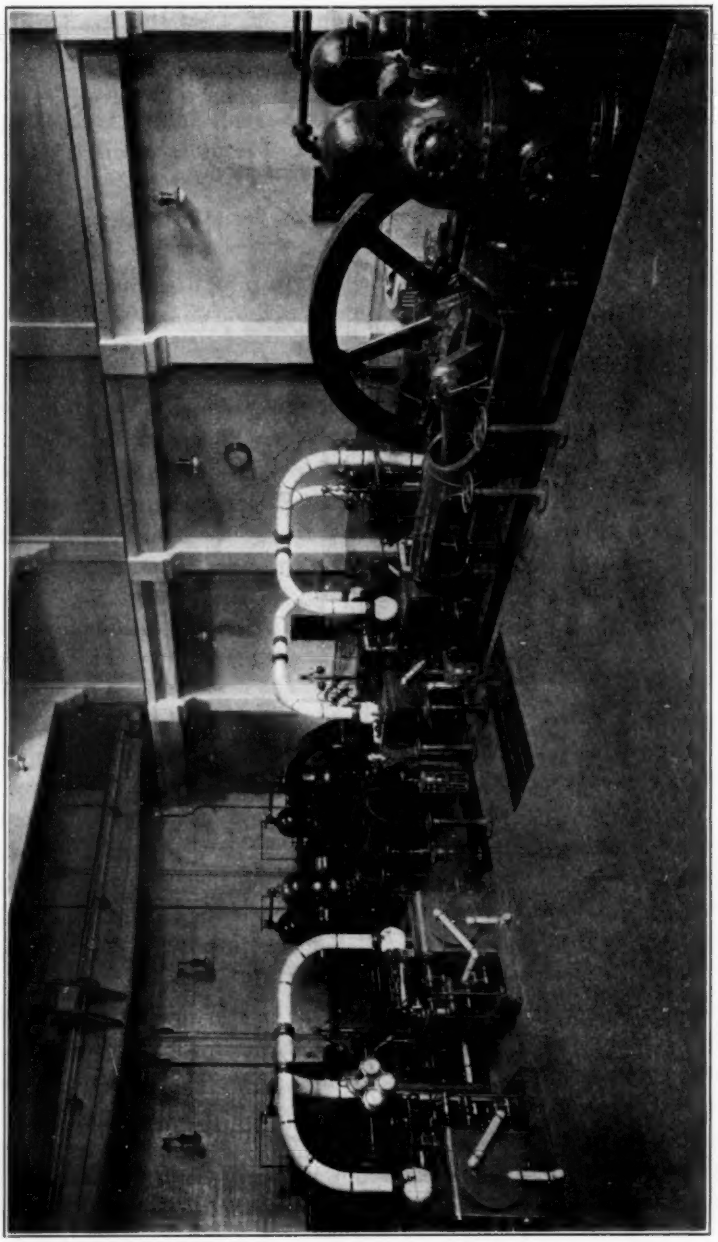


FIG. 15 INTERIOR OF STATION

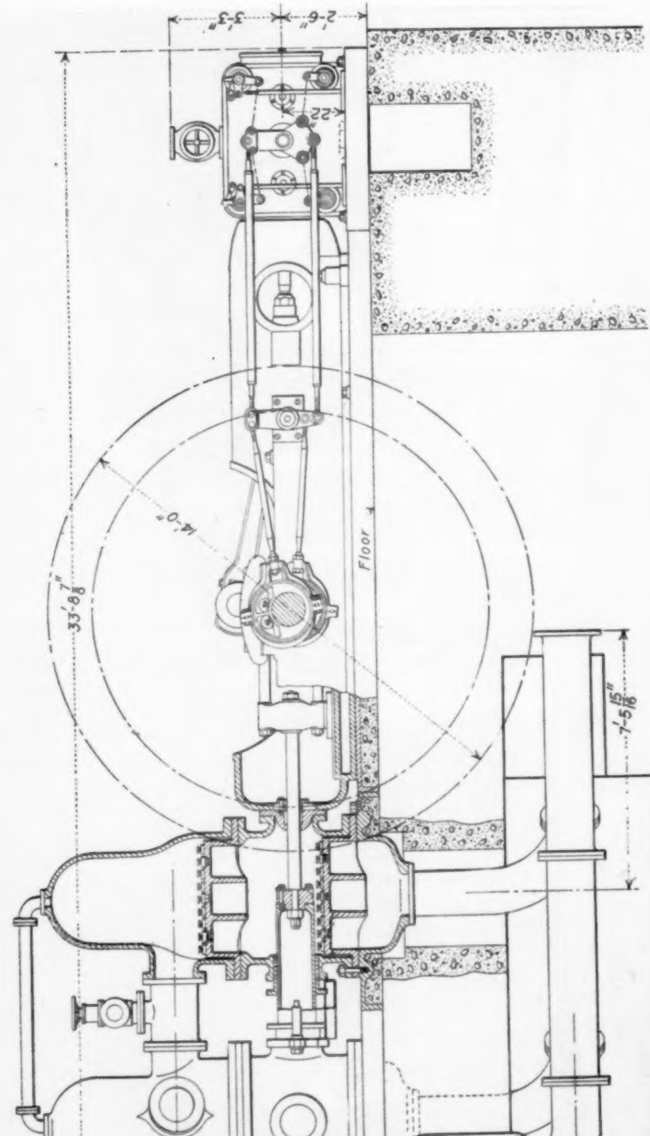



Fig. 16 Elevation of Pump

tank at the top of a 40 ft. riser, to prevent water being drawn over into the air cylinder.

65 The plant is provided with a four-motor, electric traveling crane, main hoist 20 tons capacity, auxiliary hoist 5 tons. For lighting the station and for the operation of the electrically driven auxiliaries, there are provided two non-condensing steam turbo-generators.

66 *Tests and Duty Trials.* The specifications for the pumps provided for an endurance and capacity test of 24 consecutive hours.

67 A normal load duty trial was also specified, covering a period of 12 hours; steam pressure at the throttle 125 lb. The steam consumption was to be based on the feedwater supplied to a separate boiler blanked off from all other sources of supply. The measured consumption was to include all jacket steam, but not that required for boiler feed pumps and other auxiliaries, nor separator condensation and drips from the steam piping on the boiler side of throttle.

68 In order to avoid the insertion of a water meter in either the main suction or discharge lines, a venturi meter is by-passed around a gate valve in the 30-in. city water supply. As the pump discharge can be accurately measured from the record of the pump strokes for a given period, when the slip is known, in this case the venturi meter is only used to calibrate the slip of a single pump, and therefore has a capacity only equal to one pump.

69 The duty specified is 70,000,000 ft.-lb. per 1000 lb. of dry steam, under normal operating conditions of 300 gal. per min. discharge, against a head of 250 lb., with 30 lb. pressure on the suction, and 125 lb. of steam at the throttle.

70 *Boilers.* The plant is designed for four boilers, each set singly, and each provided with a separate stack carried on structural steel supports directly over the setting. At this time only three boilers have been installed, the fourth will be added in the near future (Fig. 17).

71 The boilers are of the horizontal, inclined, straight tube type, with forged steel water legs reinforced with hollow staybolts. Each boiler contains 6800 sq. ft. of heating surface, and three 36-in. drums. All pressure parts are designed for 200 lb. working pressure. Each boiler is capable of a continuous evaporation of 45,000 lb. of water from and at 212 deg. fahr. into commercially dry steam, when burning semi-bituminous coal of approximately 14,500 B.t.u., with forced

draft not exceeding 3 in. of water in the ashpit, or the equivalent of 1 boiler h.p. from 3.77 sq. ft. of heating surface.

72 As the actual time during which the boilers will be in active service will probably average only about 100 hours per annum, the plant was designed for carrying banked fires for a large proportion of the time, with the least possible loss from radiation. The top, sides and rear of each boiler are enclosed in an air-tight steel plate casing. The hotter portions of the side walls, amounting to about 20 per cent of the area of these walls, are covered with 2 in. of magnesia blocks inside the casing. The casing plates are supported on a framework of steel angles, attached to the structural steel members supporting the boilers and stacks. These angles act also as buckstaves for the settings. Access and dusting doors are provided where necessary, which are hinged at the top and close tightly on inclined faces.

73 The boilers are set in the reverse direction from the usual method, that is, with the low end of the tubes over the furnace. The front portion of the furnace is covered with a flat fire brick arch, made of split tiles encircling and supported by the lower row of tubes. When under fire, there is presented to the furnace gases an incandescent fire brick surface, instead of the customary cool iron surface of the tubes. The first pass of the gases between the tubes is at the extreme rear of the boiler, thus providing a furnace area equal to the entire floor space occupied by the boiler inside the setting, and making it possible to force the boiler to 90 per cent over the customary rating without imperfect combustion. A special grade of fire brick was used for the furnace lining, which under a change of temperature of 3100 deg. fahr. shows an expansion or contraction of less than 0.01 in. per ft.

74 *Forced Firing.* The conditions of fire service require that the boilers shall be capable of changing from banked fires to the maximum capacity in the shortest possible time. The intervals specified were as follows: one-half rated capacity in 5 minutes; full rated capacity in 12 minutes; overload of 75 per cent in 20 minutes.

75 *Choice of Fuel.* The above conditions made imperative the use of a gaseous fuel, or fuel oil, or forced draft with coal. The possibility of the use of a combination of two of these was also considered. With gas at 90 cents per 1000 ft., this material was abandoned. The use of crude oil was given careful study. Due to the fact

that economy of operating expense was not the primary consideration, the problem was somewhat simplified.

76 As the plant is located in a wholesale district where high values of stock are common, it would have been necessary to have stored the main supply of oil underground. Ample space for this purpose existed in the wide water front street about 200 ft. from the station. On the other hand, experience has demonstrated that oil should be fed to the burners only by gravity from an overhead storage of suitable capacity. Experience has also shown that in spite of all precautions, an oil storage tank will in all probability, sooner or later, take fire. It was, of course, possible to design a fireproof barrier which would prevent the fire doing damage to adjoining property. It was believed, however, that in a plant designed for fighting conflagrations, the possibility of an oil fire, with its huge volumes of dense black smoke, would not tend to popularize the system, even if the fire proved to be entirely harmless. In spite of its recognized advantages in ease of handling and quick firing, the use of fuel oil was therefore definitely abandoned, and coal was adopted. The introduction of natural gas from the West Virginia fields has been under consideration for some years, and if this should be accomplished, it would make an admirable fuel for the purpose, either alone or as supplementary to the coal furnaces.

77 *Automatic Mechanical Stokers.* The use of coal made necessary also the use of forced draft. In addition, it was desirable that during the periods of banked fires, there should be maintained a full bed of ignited coal, requiring only the air blast to force the fire to the highest rate of combustion. The underfeed type of stoker seemed to meet these conditions very satisfactorily. A large body of coal, approximately 1000 lb., can be carried in each furnace, a part of this coal incandescent, a part coked and a part in the process of coking. This represents about 14,500,000 B.t.u. in storage ready for use on a few minutes' notice. As a further heat storage, the steam pressure, which is ordinarily carried at about 150 lb., can be raised to 200 lb., immediately upon the receipt of an alarm. As the normal operating pressure is only from 125 to 150 lb. at the pump throttles, by the time the hose companies can reach the hydrants and attach the hose, and put a sufficient draft on the pumps to pull the steam down to normal, the furnace fires will be ready to respond to any demand.

78 Each boiler is equipped with four underfeed stokers, and each stoker unit is capable of burning efficiently, without smoke, 1000 lb.

of coal per hour. For the four boilers a duplicate blower equipment is installed, each consisting of a full housed steel plate fan of sufficient capacity to operate the four boilers at 75 per cent overload. Each blower is driven by a direct-coupled vertical steam engine, and is connected to the main air pipe line, and by means of dampers either blower can be used to serve any or all the boilers.

79 To regulate the supply of both fuel and air in the proportion required for complete combustion under all rates of firing, the stokers are provided with automatic regulators actuated by the boiler pressure. Provision is also made for hand regulation, so that it is possible to anticipate sudden demands for steam upon receipt of an alarm, and also for decreasing the supply of both fuel and air when the demand for forced firing has passed.

80 *Stacks.* Each boiler is provided with a separate steel stack located directly over the boiler which it serves. Each stack is 72 in. diameter by 125 ft. above the boiler room floor, and is carried by structural steel framing from the boiler foundations. The stacks are unlined.

81 *Coal Handling and Storage.* Semi-bituminous run of mine coal is used almost entirely for steaming purposes in the vicinity of Baltimore. In this instance the coal is delivered in carts, the total annual consumption being too small to justify mechanical handling under existing conditions. The coal is dumped on a grating over an opening in the sidewalk, the mesh being 4 in. sq. Large lumps are broken up with a maul, as with the type of stoker in use there is no necessity for the coal to be crushed to smaller sizes. Below the grating there is a dumping hopper which receives the coal and delivers it to the lower run of a bucket elevator. The elevator is a double strand link belt with V-shaped buckets, 16 in. by 15 in., dumping by gravity. The outfit has a capacity of 25 tons per hour with uniform feed.

82 The coal is delivered into reinforced-concrete bunkers with inclined bottoms, located directly over the fireroom. The bunkers have a capacity of 150 tons without trimming, or sufficient to operate the entire plant at its maximum capacity for 30 hours. Coal can also be delivered from carts directly on to the boiler room floor, so that the operation of the plant is not dependent upon the coal handling equipment nor the storage supply.

83 *Boiler Feedwater.* Boiler feedwater is normally supplied from the city mains under 30 to 50 lb. pressure, sufficient to deliver

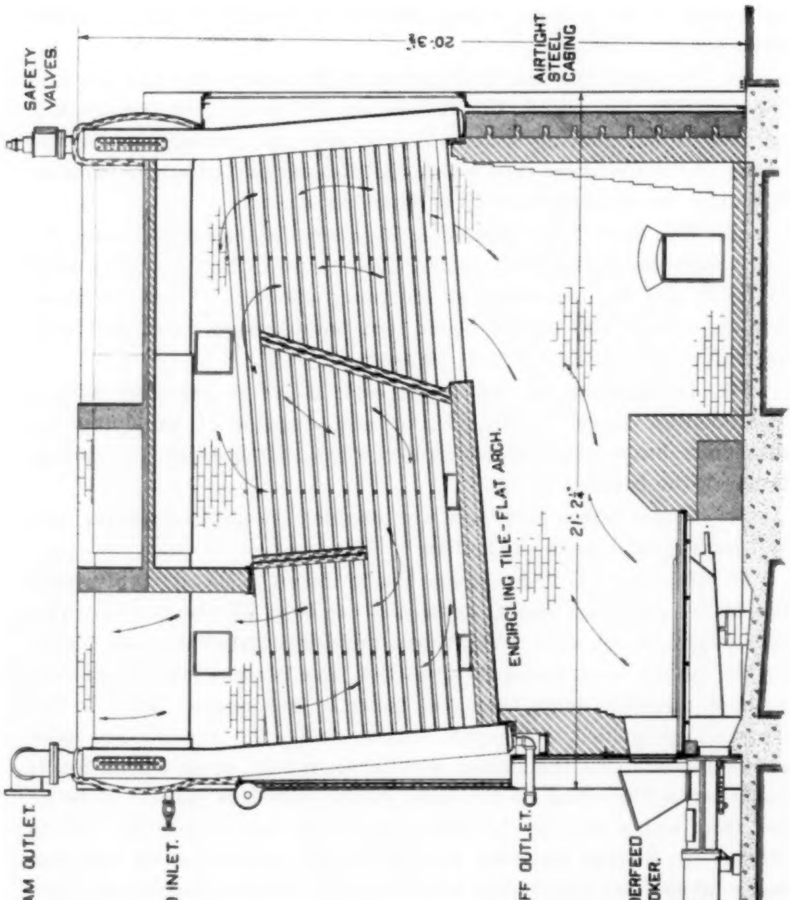



FIG. 17 SECTION OF BOILER

to the open heaters in the gallery of the engine room, without pumping. In the event of an interruption to the city supply, the feed-water can be taken from a reserve supply stored in steel tanks in the basement under the boilers. For lifting from the storage tanks and delivering to the heaters a duplicate set of low-service duplex steam pumps is provided.

84 To avoid the loss of efficiency in the boilers due to scale, and the necessity for taking the boilers out of service for considerable periods while removing scale, a double unit hot process purifier and heater of the Cochrane type was installed, each half of ample capacity to handle the consumption of the entire plant.

85 The heaters are located in a gallery in the engine room, directly over the boiler feed pumps, thereby providing a gravity head of 25 ft. for the hot water to the pump suction. There are three duplex, outside end packed boiler feed pumps, brass fitted and with pot valves, designed for 300 lb. pressure.

86 In addition to the above, each boiler is provided with a Metropolitan Model O No. 7½ injector, capable of supplying the maximum evaporation of the boiler when lifting from the storage tanks in the basement.

87 Foster excess governors are provided on the feed pumps, and Williams regulators on the boilers.

88 *Piping.* A 12-in. steam header forms a closed ring around the plant, with long radius expansion bends at all changes in direction (Figs. 18, 19, 20). A sufficient number of gate valves are placed in the header to sectionalize it, so that any portion may be cut out without disabling more than one boiler or one pump. Pipe is full weight, lap welded, soft open hearth steel. To provide an independent header for the station auxiliaries, a 6-in. cross connection is made across the center of the main header, which is capable of being fed from either side of the main header, in case of accident to the other. No fittings whatever are used in the main line, all branches being taken from interlocked welded necks. Boiler branches are provided with non-return valves at the boiler nozzles and gates at the header end. Van Stone flanges are provided for connections to the valves and receivers, which are located so as to avoid as far as possible the necessity for any additional joints in the line. Wrought steel receiver type separators are installed at the low points on each side of the header.

89 The exhaust system is extremely simple, a multi-port back

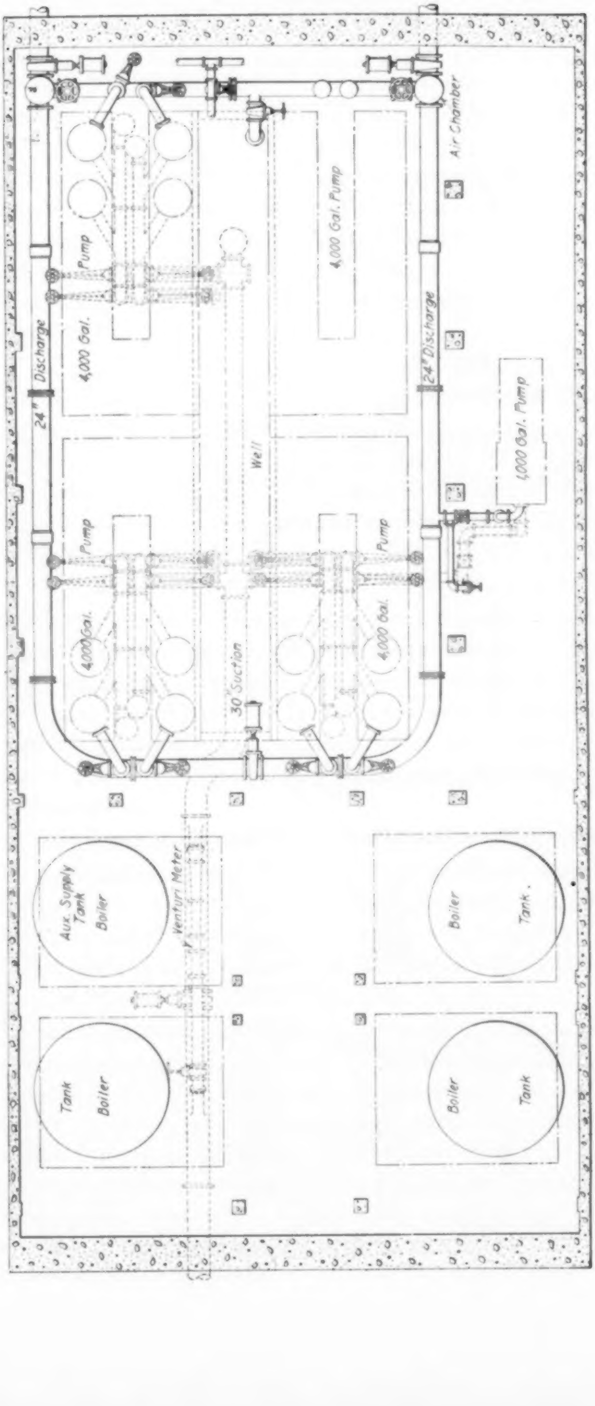



FIG. 18 PLAN OF STATION, SHOWING WATER PIPING

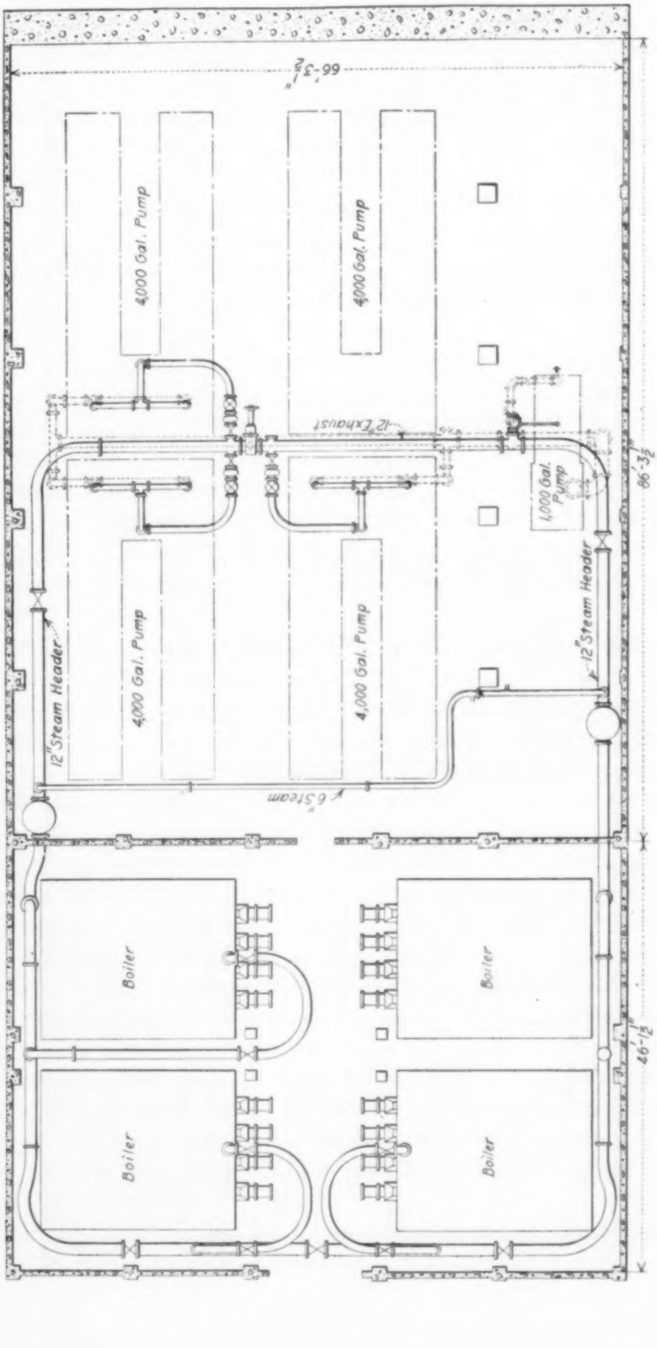



FIG. 19 PLAN OF STATION, SHOWING STEAM PIPING

pressure valve on the 18-in. riser serves to turn all the exhaust into the heaters at light loads, and provides a direct path to the atmosphere when the steam is in excess of the heater requirements.

90 *Building.* The site of the pumping station is a lot 69 ft. front by 137 ft. deep, running back to a 16 ft. alley. The property immediately adjoining on the two sides is occupied by warehouses carrying more or less inflammable stocks. On the opposite side of the alley the same conditions exist. All foundations for the pumping station and machinery were designed to rest on caissons carried to the gravel, so that it was necessary to underpin the walls of both the adjoining buildings.

91 All structural portions of the building, including columns, girders, beams, floor and roof slabs, and all walls except the front, are of reinforced concrete. The adjoining buildings on the sides are from 10 to 20 ft. higher than the station roof, so the roof girders, beams and slabs were designed to withstand the shock of falling walls in case of fire. The side walls are not less than 8 in. thick in any part, and have no openings whatever. The rear wall is the same, except that there is one door at the level of the boiler room floor.

92 For access to the men's quarters an automatic push-button electric elevator is installed, in addition to the stairway. Standard brass sliding poles are also provided for quick response to an alarm. The quarters are located over the front of the engine room, and include a dormitory, dressing room, bath, toilet and reading rooms. In addition, a private bedroom, bath, parlor and office are provided for the chief engineer.

93 In addition to the fireproof construction of the building, further fire fighting equipment for the protection of the station consists of a water curtain for the exposed front and rear, and two 8-in. standpipes, to be fitted with monitor nozzles. A dangerous fire in the immediate vicinity of the station could thus be effectively fought from the roof as well as from the ground.

94 *Signaling System.* In addition to the regular fire alarm circuit, a separate telephone circuit runs to the pumping station from fire alarm headquarters, fire department headquarters, and the chief's night quarters. This circuit connects to contacts for portable telephones in each fire-alarm box in the high-pressure district. In addition to the regular Morse key and sounder there are contacts for a telephone connection in each box, over the fire alarm circuit. Finally there is available the regular public telephone service.

## CONSTRUCTION COSTS

| | | | |
|---|---|---|---|
| **Portable Equipment** | | | |
| 2 automobile hose wagons at $5000 | | | $10,000 |
| 8000 ft. 3 in. hose at $1 | | | 8,000 |
| 30 portable heads and regulators at $385 | | | 11,550 |
| Total | | | $29,550 |
| **Pipe System** | | | |
| Material delivered Baltimore | | | |
| Hydrants, 226 at $100 | $22,600 | | |
| 8 in. pipe, 7137 ft. at $2.35 | 16,700 | | |
| 10 in. pipe, 28,229 ft. at $3.10 | 87,700 | | |
| 16 in. pipe, 17,052 ft. at $5.25 | 89,600 | | |
| 24 in. pipe, 1275 ft. at $10 | 12,750 | | |
| 8 in. gate valves, 6 at $100 | 600 | | |
| 10 in. gate valves, 193 at $130 | 25,000 | | |
| 16 in. gate valves, 90 at $210 | 18,900 | | |
| 18 in. gate valves, 2 at $300 | 600 | | |
| 24 in. gate valves, 3 at $1,000 | 3,000 | | |
| Air and relief valves | 200 | | |
| Low pressure gates, 2–30 in | 500 | | |
| Suction pipe, 400 ft. cast iron, 30 in., at $4 | 1,600 | | |
| Steel air chambers, 2–30 in., at $500 | 1,000 | | |
| Venturi meter | 500 | | |
| Cast steel specials | 17,500 | | |
| | | $298,750 | |
| **Installation** | | | |
| Laying pipe, including placing valves, fittings, hydrants, etc. | | | |
| 8 in. pipe, 7,137 ft. at $0.70 | $4,996 | | |
| 10 in. pipe, 28,229 ft. at 0.75 | 21,200 | | |
| 16 in. pipe, 17,052 ft. at 1.15 | 19,600 | | |
| 24 in. pipe, 1,275 ft. at 1.75 | 2,230 | | |
| Pump connections in station | 6,000 | | |
| Laying 30 in c. i. suction | 3,400 | | |
| Tapping 40 in. main | 1,500 | | |
| Concrete valve boxes, 293 at $30 | 8,790 | | |
| Excavation, back filling and rubble paving | | | |
| 41,318 ft. open trench, at $3.84 | 158,600 | | |
| 12,375 ft. tunnel, at $4.08 | 50,400 | | |
| Improved paving, 6650 sq. yd., at $1.50 | 10,000 | | |
| Superintendence, use of tools, etc | 50,000 | | |
| | | $336,716 | |
| | | | $635,466 |

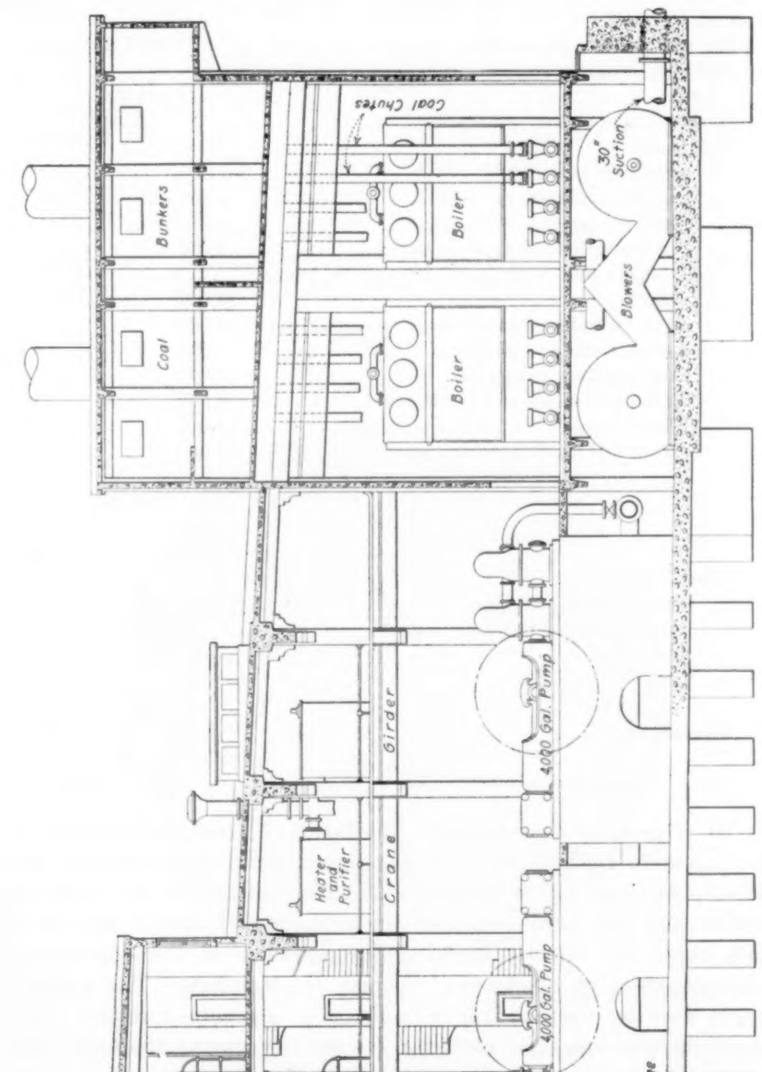



FIG. 20 SECTIONAL ELEVATION OF STATION

| | | | |
|---|---|---|---|
| Pumping Station | | | |
| Site and preliminary work | | $37,730 | |
| Building, including machinery foundations, and men's quarters | | 124,800 | |
| Harbor intake and screen chamber | | 10,000 | |
| Equipment | | | |
| Four 4000 gal. pumps | $82,000 | | |
| One 1000 gal. pump | 3,500 | | |
| Auxiliary pumps | 4,250 | | |
| Feedwater heaters and purifiers | 4,750 | | |
| 4 boilers and settings, 27,200 sq. ft. heating surface | 33,000 | | |
| 16 underfeed stokers, blowers, air piping, etc. | 18,000 | | |
| 4 steel stacks and supports | 8,000 | | |
| Coal handling apparatus | 7,000 | | |
| Turbo-generators and switchboard | 4,500 | | |
| Electric crane | 4,000 | | |
| Steam and auxiliary water piping | 30,000 | | |
| | | $199,000 | |
| | | | 371,530 |
| Miscellaneous | | | |
| Signal system, cables, etc. | $1,500 | | |
| Furnishings for men's quarters | 500 | | |
| Incidentals | 5,000 | | |
| | | | 7,000 |
| Engineering | | | 50,000 |
| Total cost of construction | | | $1,093,546 |

95 *Operating Department.* The Board of Fire Commissioners is the executive head of the operating department, acting through the department chief. The operation and maintenance of the pumping station and the maintenance of the pipe lines, hydrants and portable heads are directly under the supervision of the department superintendent of machinery, Thomas H. Meushaw. The maintenance work for the pipe line is in charge of a general foreman, John Rudolph, who was chief inspector for the city during the early part of the installation, and general foreman for the contractor during the latter portion of the work.

96 The operating force at the pumping station is in charge of a resident engineer and five assistant engineers, with four stokers and two general assistants, all organized as a fire company. Two additional stokers have been recommended by the superintendent of ma-

chinery, and will probably be added in the near future, making a total station operating force of fourteen men. Of these an engineer and stoker are on active duty at all times, with a four-hour watch. Immediately upon the receipt of an alarm all hands report on the operating floor. As a pressure of 150 lb. is maintained during the standby period, orders have been issued that in the event of the pumps automatically speeding up without an alarm, the plant shall be shut down immediately. This course is taken to avoid water damage, should a break occur. Up to this time no occasion has arisen to require the execution of this order, however.

## TESTS

97 *Opening Demonstration.* The system was formally placed in

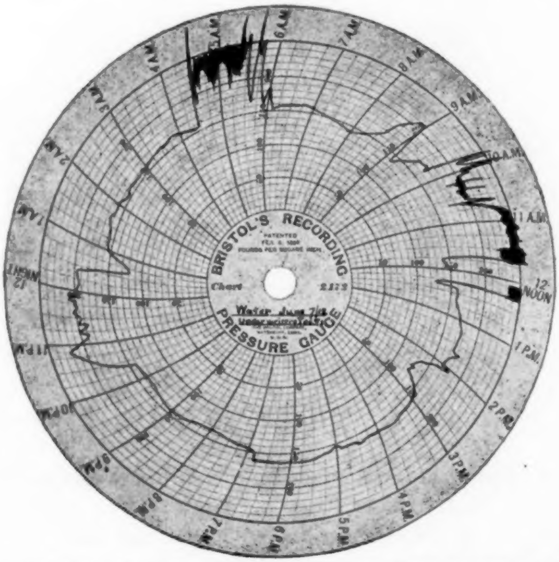



FIG. 21 CURVE OF WATER PRESSURE DURING UNDERWRITERS' TESTS

operation on May 20, 1912, with a public demonstration on the Court House Plaza. At first twenty-four 1¾-in. streams were used from single hose lines held by tripods. Later seven 2½-in. streams were thrown, from monitor nozzles on the wagons. About 13,000 gal. per min. were delivered.

98 *Underwriters' Tests.* A readiness test was made June 7, readings being taken by a stop-watch.

4.30.00 a.m. Fire alarm box pulled; one engineer and one stoker on duty at station.

4.31.00 a.m. Chief engineer with four additional engineers and three additional stokers had responded from sleeping quarters.

4.31.15 a.m. Two large pumps started up, pressure increased from 150 to 190 lb.

4.33.00 a.m. Pressure 280 lb. Hose company turned water on to two 3-in. hose lines siamesed into one 2½-in. monitor nozzle.

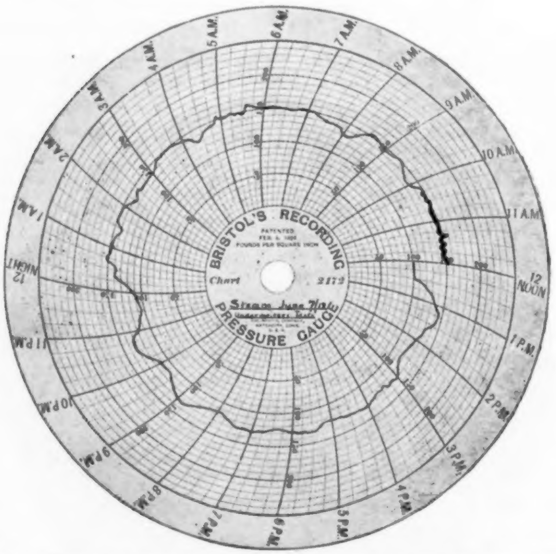



FIG. 22 CURVE OF STEAM PRESSURE DURING UNDERWRITERS' TESTS

4.34.00 a.m. Discharge was 1050 gal. per min.

4.37.40 a.m. Two additional 2½-in. nozzles in service, total discharge 4000 gal. per min.

99 A second company was then ordered into service at another point, with three 2½-in. nozzles, bringing the total discharge up to 7100 gal. per min. See water and steam charts from Bristol gage at station (Figs. 21 and 22).

100 A general performance test was made June 7, from 9.45 to 10.15 a.m. (see Bristol charts). Pumps were started and stopped, and hydrants were opened and closed as rapidly as possible to test

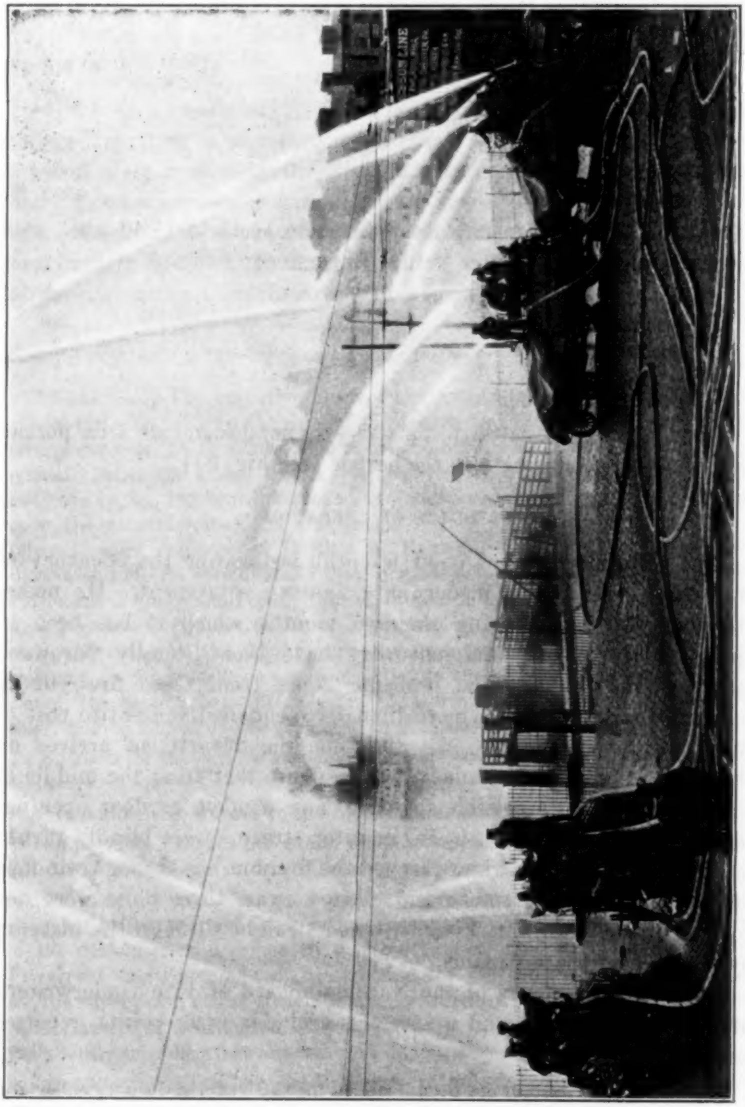


Fig. 23 View of Streams during Underwriters' Tests

the ability of the pumps and governors to take care of abnormal operating conditions.

101 Individual pump tests were made the same date, from 10.15 to 10.45 a.m., with the following results:

| | |
|---|---|
| Steam at throttle, lb | 146 |
| Water pressure, discharge, lb | 199 |
| Water pressure, injection, lb | 41 |
| Water pressure, net, lb | 158 |
| Revolutions per minute | 55 |
| Discharge, gal. per min | 4592 |

102 A general capacity test was made 10.45 to 11.40 a.m., with the following results (three pumps in service):

| | |
|---|---|
| Steam at throttle (average), lb | 145 |
| Water, discharge, lb | 240 |
| Water, injection | ? |
| Water, net | ? |
| Discharge, gal. per min | 12,770 |

103 During the latter test six $2\frac{1}{2}$-in. nozzles and six 2-in. nozzles were in use, discharging into the harbor (see Fig. 23).

### RESULTS OF OPERATION

104 The department chief is enthusiastic over the efficiency of the entire system as a modern fire fighting equipment. He makes the statement that during the two months which it has been in operation, three bad incipient fires have been literally "drowned out" by the system. The probable losses from these fires, under former conditions, would more than have equalled the entire cost of the high-pressure service. By the time the department arrived on the scene at one fire, the smoke was so dense that from the middle of the street it was impossible to locate any window or door openings in the building. Three $2\frac{1}{2}$-in. monitor streams were blindly turned on the building. The water reached the fire, but not through windows or doors. When the smoke had cleared away, three holes were discovered in the 18-in. brick walls, bored straight through the masonry by the high-pressure streams.

105 The engineers of the National Board of Fire Underwriters, after a thorough test and a special search for weak points, reported as follows:

The distributing system has been installed for two years and shows no signs of deterioration. The slight leakage, absence of electrolytic action and total freedom from breaks or other troubles appear to justify the departure from the usual design of such systems. . . . The valve and hydrant distribution is

excellent, and the pipe sizes and gridironing are sufficient to enable a good concentration of flow without serious loss of pressure. . . . The separate hydrant head permits the use of regulator valves permanently attached, giving excellent control of the pressure on hose lines. The hydrant head under test showed sufficiently low friction loss. . . . The operation of the pumping plant is prompt and reliable.

106 As a result of the installation of the high-pressure system, the underwriters have announced a rebate of 5 cents in the insurance rate on all property in the district covered. While no exact summary has been made of the aggregate saving which will result from this reduction, it is roughly estimated that the amount will be approximately $40,000 per annum, which will be increased almost in direct proportion with the extension of the mains.

## ORGANIZATION

*Construction.* The executive head of the project is the Board of Fire Commissioners. To act with the chief of the department the board appointed a consulting engineer, D. B. Banks, who in collaboration with the writer, designed the system. After the general plans had been drawn, but before the construction had been begun, the board employed two additional consulting engineers to pass upon the general features of the design. R. C. Carpenter and Frederick H. Wagner, chief engineer of Bartlett & Hayward Company, Baltimore, were chosen, and after a careful study these engineers approved the general plans and the details as far as completed. The architectural features of the station were designed by Henry Brauns, the veteran power plant architect of Baltimore. General supervision of the construction was exercised by Wm. McCallister, Jr., assistant to the consulting engineer. Before the construction work was completed, the department chief, George W. Horton, was retired, and the deputy chief, August Emerich, was promoted to the head of the department.

Contracts for the various elements of the system were awarded to the following builders and contractors:

Automobile hose wagons to the Mack Manufacturing Company.

The portable hydrant heads and regulators, and the main pump governors to the Ross Valve Manufacturing Company, of Troy, N. Y.

The hydrants and high pressure water pipe lines to the Pittsburgh Valve Foundry & Construction Company of Pittsburgh. Many of the working details of the system were designed by J. Roy Tanner, chief engineer, and Charles Fitzgerald, superintendent of construction for the contractor. A subcontract for the supervision of the trenching was awarded by the general pipe contractor to E. Saxton, of Washington, D. C., one of the most experienced contractors in the vicinity on subsurface structures in city streets.

Pumps to the Allis-Chalmers Company, of Milwaukee, although this concern was not the lowest bidder. A subcontract for the 1000-gal. direct acting pump and the boiler feed pumps was awarded to the Epping Carpenter Company, of Pittsburgh.

Boilers to the Edge Moor Iron Company of Wilmington, Del. A subcontract for the stokers, blowers and regulators was awarded to the Underfeed Stoker Company of America, Chicago.

The steam and auxiliary water piping in the pumping station to the Crook-Kries Company, of Baltimore.

The station building and harbot intake to the B. F. Bennet Building Company, of Baltimore.

The signal system was installed by the department force.

No. 1393 c

# ALLOWABLE HEIGHTS AND AREAS FOR FACTORY BUILDINGS

By Ira H. Woolson, New York

Member of the Society

In the design of factory buildings, one of the vital features tending to control the spread of fire is a judicious limitation of height and area. It is self-evident that whatever restricts a fire reduces the life hazard. Owing to the supreme importance of these two subjects, a person contemplating the erection of a building of this class should give careful consideration to the history of fires in such buildings, and the experience gained in fighting them. The question is more acute in this class of buildings than in any other because of the fire hazard which exists in them, and the economic advantages due to reduced costs in construction and supervision, when several large areas are housed under a single roof. Just where to draw the line so as to produce reasonable safety without prejudice to building investments is the problem.

2 Factory buildings of excessive heights or areas have long been recognized by underwriting organizations as a grave danger to life and property, owing to the difficulty of controlling fires in them. They have for years urged limitations which have been freely ignored by ambitious architects and factory owners, because the suggested restrictions were considered unreasonably drastic. The evidence produced in this paper strongly supports the limitations which were advocated.

3 It is logical to assume that the men best fitted to determine safe limits of heights and areas are the men who have made a life work of combating fires under all conditions of weather and hazard. With this idea in mind, the writer communicated with all the fire marshals and fire chiefs in the United States representing cities of over 20,000 population. A set of eight questions and a letter of explanation were sent to each. Fire chiefs as a class are not good technical correspondents, therefore it was not surprising that only

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

one-third of the men addressed responded to the appeal. However, replies were received from 117 representative cities well distributed as to size and geographical location. These have been summarized and form the basis of this paper. A few of the replies indicated a misunderstanding of the questions, and these were discarded. The questions were as follows:

1 What should be the greatest height allowed for manufacturing or warehouse buildings *without sprinkler equipment?*

Brick and joist construction..........Height in ft.—— or No. of Stories——
Fireproof construction................Height in ft.—— or No. of Stories——

2 Take the same question as No. 1, but assume the buildings to be fully equipped *with automatic sprinklers.* What height would you approve?

Brick and joist construction..........Height in ft.—— or No. of Stories——
Fireproof construction................Height in ft.—— or No. of Stories——

3 What should be the greatest floor area allowed in the same class of buildings *without sprinkler equipment?*

Brick and joist construction Area in sq. ft.—— or Width—— ft Length—— ft.
Fireproof construction......Area in sq. ft.—— or Width—— ft. Length—— ft.

4 If the same buildings were fully equipped *with automatic sprinklers* what area would you approve?

Brick and joist construction Area in sq. ft.—— or Width—— ft. Length—— ft.
Fireproof construction .....Area in sq. ft.—— or Width—— ft. Length—— ft.

4 Each building was assumed as a good one of its class, with enclosed stairways and elevator shafts; and the chiefs were requested to base their answers upon experience in fighting fires in the class of buildings described, and to assume restrictions which would afford a reasonable chance of controlling a fire on any floor.

TABLE 1 GENERAL AVERAGE OF 99 TO 111 REPLIES RECEIVED FROM ALL CLASSES OF CITIES[1]

| Type of Building | Stories in Height | Area between Fire Walls in Sq. Ft. |
|---|---|---|
| Non-fireproof, not sprinklered | 3.1 | 6,300 |
| Fireproof, not sprinklered | 4.9 | 12,300 |
| Non-fireproof, sprinklered | 4.6 | 12,800 |
| Fireproof, sprinklered | 7.0 | 27,100 |

Average story height was 12 to 13 ft.

5 Naturally, and quite properly, the replies reflect the local conditions, such as the efficiency of the fire department, the water pressure, the combustibility of the goods being manufactured, the

[1]The variation in the number of replies (averaged) resulted from some incomplete answers.

number of sprinkler equipments in service, and the degree of congestion among the buildings. However, all conditions were represented, and the summary of so large a number of opinions should indicate fairly well the average condition throughout the country. (See Table 1.)

6 The answers regarding allowable heights were much more uniform than those relative to area. It is significant that 83 per cent of the replies would limit the height of a fireproof sprinklered factory building to less than ten stories. The opinions in reference to height of the other classes of buildings were exceedingly uniform, and consistently low.

7 Replies as to permissible areas in sprinklered buildings were widely divergent, but for the unsprinklered classes they were more uniform than would naturally be expected considering the great diversity of conditions under which they were prepared.

8 It is evident from the figures given, that the fire chiefs have no settled policy among themselves as to the credit that should be given to an automatic sprinkler equipment as a fire extinguishing device. A few enthusiasts would permit unlimited area in a sprinklered building, while on the other hand a considerable number would give very little or no increase, when sprinklers are installed. Two chiefs stated that their unfortunate experiences with sprinklers had caused them to lose faith in their reliability. As a whole, however, they are strongly in favor of sprinklers and are inclined to permit overgenerous areas in buildings so equipped.

9 In order that the replies may be intelligently interpreted they have been separated into three groups, Tables 2, 3, 4, according to size of the city represented, and each group has been analyzed to show the character of the answers given to each question.

10 In the cases referred to by an asterisk, where no limits to areas were given, they were not included in the averages, but were counted in the columns giving the number of answers above the average. In each group it will be noted that about the same number of men gave high answers to all questions, the proportion being one-quarter to one-half of the number in the group. The uniformity of height limits, and the lack of it in the area limits, is very apparent in all groups. It will be noted that the largest area values are given in Groups I and II, comprising the smaller cities. This is significant, and needs explanation.

11 Occasionally the fire chief of a small city has experience which would abundantly qualify him to estimate properly the merits of fire-

proof construction and sprinkler equipments; more often, however, his city has meager protection of this kind, and consequently he has little

TABLE 2 GROUP I SUMMARY OF ANSWERS FROM 52 CITIES WITH A POPULATION OF 20,000 TO 50,000

| Type of Building | Stories in Height | | | Answers above Average | Area in Square Feet | | | Answers above Average |
|---|---|---|---|---|---|---|---|---|
| | Average | Max. | Min. | | Average | Max. | Min. | |
| Non-fireproof, not sprinklered | 2.8 | 6 | 1 | 13 | 6,000 | 20,000 | 1,150 | 15 |
| Fireproof, not sprinklered | 4.4 | 10 | 2 | 24 | 12,600 | 60,000 | 1,150 | 15 |
| Non-fireproof, sprinklered | 4.1 | 8 | 2 | 17 | 12,300 | *60,000 | 3,000 | 17 |
| Fireproof, sprinklered | 6.3 | 12 | 3 | 18 | 27,300 | *180,000 | 5,000 | 20 |

* Four votes received in favor of "no limit to area" in this class.

TABLE 3 GROUP II SUMMARY OF ANSWERS FROM 23 CITIES WITH A POPULATION OF 50,000 TO 100,000

| Type of Building | Stories in Height | | | Answers above Average | Area in Square Feet | | | Answers above Average |
|---|---|---|---|---|---|---|---|---|
| | Average | Max. | Min. | | Average | Max | Min. | |
| Non-fireproof, not sprinklered | 3.2 | 6 | 1 | 8 | 8,300 | 40,000 | 2,500 | 5 |
| Fireproof, not sprinklered | 5.2 | 10 | 1 | 6 | 14,800 | 60,000 | 2,400 | 4 |
| Non-fireproof, sprinklered | 4.8 | 10 | 3 | 5 | 16,300 | 75,000 | 1,500 | 5 |
| Fireproof, sprinklered | 7.7 | 20 | 4 | 5 | 36,300 | 200,000 | 4,000 | 5 |

TABLE 4 GROUP III SUMMARY OF ANSWERS FROM 36 CITIES WITH A POPULATION OF 100,000 AND OVER

| Type of Building | Stories in Height | | | Answers above Average | Area in Square Feet | | | Answers above Average |
|---|---|---|---|---|---|---|---|---|
| | Average | Max. | Min. | | Average | Max. | Min. | |
| Non-fireproof, not sprinklered | 3.5 | 7 | 1 | 17 | 5,400 | 10,000 | 900 | 15 |
| Fireproof, not sprinklered | 5.3 | 9 | 2 | 18 | 9,800 | 22,500 | 2,400 | 10 |
| Non-fireproof, sprinklered | 5.0 | 10 | 3 | 15 | 11,300 | 22,500 | 900 | 13 |
| Fireproof, sprinklered | 7.5 | 12 | 4 | 16 | 19,400 | *80,000 | 2,500 | 9 |

* Two votes received in favor of "no limit to area" in this class.

opportunity to judge of their efficiency, and it is not strange that he should be a bit extravagant in the credit he would give them.

12 The most rigid restrictions on area are found in Group III embracing the large cities. As fireproof construction and sprinkler equipments are common in most of our large cities, it is reasonable to assume that the fire chiefs of such cities would have had more experience with such methods of protection, and be better able to decide what increase should be given in the size of a building when such protection is provided, than their less experienced fellow officers in smaller towns. It is thought quite proper to assume their figures are more nearly correct and should be given the most weight.

13 Significant evidence in support of this argument is found in the fact that four chiefs who give no limit to areas in non-fireproof and fireproof sprinklered buildings are located in cities having a population of less than 50,000 in which there are few fireproof factory buildings or sprinkler equipments. On the other hand only two chiefs, in cities over 100,000 population, suggest a "no limit area" in a fireproof sprinklered building, and none approves such areas for non-fireproof buildings.

TABLE 5 ALLOWABLE HEIGHTS AND AREAS IN FACTORY BUILDINGS

| Type of Building | Stories in Height | Area between Fire Walls in Sq. Ft. |
|---|---|---|
| Brick and joist construction, not sprinklered | 3 | 6,000 |
| Fireproof construction, not sprinklered | 5 | 10,000 |
| Brick and joist construction, sprinklered | 5 | 13,000 |
| Fireproof construction, sprinklered | 8 | 20,000 |

TABLE 6 AREAS IN FACTORY BUILDINGS

AVERAGE OF THE REPLIES OF 50 FIRE CHIEFS SELECTED FROM 117, THE TOTAL NUMBER RECEIVED AS BEST QUALIFIED BY TRAINING AND EXPERIENCE TO PASS JUDGMENT ON THE QUESTIONS INVOLVED

| Type of Building | Stories in Height | Area between Fire Walls in Sq. Ft. |
|---|---|---|
| Brick and joist construction, not sprinklered | 3.2 | 5,200 |
| Fireproof construction, not sprinklered | 5.3 | 9,300 |
| Brick and joist construction, sprinklered | 4.8 | 10,500 |
| Fireproof construction, sprinklered | 7.5 | 21,600 |

14 With these thoughts in view, Table 1 has been changed somewhat to be more in accord with the weight of evidence. It is believed, therefore, that Table 5 represents more correctly the consensus of opinion among the fire chiefs of the country best qualified

to judge as to what should be the proper limits of height and area for factory buildings.

15 These values might be increased somewhat under the influence of especially favorable local conditions, as previously explained, but the writer submits that as they represent the average deliberate judgment of such a large body of men, so well qualified to estimate the hazard which the values involve, they should be given careful consideration, and should be increased only with the utmost caution.

In selecting the above replies, attention was specifically given not only to the personality of the fire chief, but also to the character and number of factory buildings in his city, and the probability of his having experience with both fireproof construction and sprinkler equipment.

The chiefs selected were distributed according to size of cities as follows: 32 from cities with a population of over 100,000; 14 from cities with a population of 50,000 to 100,000; 4 from cities with a population of 20,000 to 50,000.

It will be noted that the figures in this table of actual averages compare very closely with those given in Table 5 which was compiled by a somewhat arbitrary method in an effort to bring out the same facts.

## EXTRACTS FROM FIRE CHIEFS' LETTERS

16 The following extracts from letters received from different fire chiefs in connection with this investigation may be of interest as indicating their attitude of mind in relation to the questions asked:

"In my opinion, from a fire-fighting standpoint, *no building* should be built over eight stories."

"In our city there is room to grow on the ground without building high in the air. It is almost impossible for a public fire department to fight a fire from the outside above 75 ft."

"The figures given mean that every 66 ft. by 66 ft. should have a brick wall through length of building with Underwriters' doors, same to be double. As for width, in no case over 66 ft. wide; with solid wall, same to reach above roof at least 6 ft. *Build on ground not in air.*"

"A building 8 or 10 stories high, out in the open where it can be attacked from all sides should be handled very readily by a modern equipped fire department."

"I think that a factory should never be more than four stories high. I almost feel that there is no such thing as fireproof construction from my own experience. I know that it is possible to store enough material in any building to burn it. I am very much in favor of dividing rooms in factories with fire-resisting walls, provided with automatic fire doors."

"While fireproof construction is the best, it is the contents placed therein that is the hazard to life and property. Buildings should not be constructed

to a greater height than can be reached by fire department ladders; 85 ft. to upper windows."

"In my opinion no warehouse building ought to be over one story in height. In regard to manufacturing buildings, I will say that I do not approve of any of these buildings being over three stories in height. If they want room, let them build in length and not so high; that is just what makes such bad fires. These buildings have all kinds of combustible material in them and they are sure to jump to another building if they are four or five stories in height."

"It is my opinion that all buildings for manufacturing and warehouses should be sprinklered, and not built higher than what the water supply will furnish and cover."

"Do not think any fire department can successfully fight a large fire over six stories high, and ten stories allowed only when there are two sources of water supply with good pressure."

"Area of sprinklered and unsprinklered buildings should be about the same, on account of increase in height allowed for fireproof buildings."

"All buildings of character named should be sprinklered."

"Joisted brick construction should not be allowed without sprinklers."

"I think a good sprinkler system is one of the best fire preventions that has been invented in a great many years, and if kept up properly, it is pretty hard for fires to get away."

"If I had my way I would not allow any manufacturing plant to do business until it were properly sprinklered. It does things when they should be done."

"My experience with the 28 factories in this city has been that the sprinkler systems are out of order much of the time. Not looked after properly."

"This department has had no unfortunate experience with the sprinkler system, but, I do not feel inclined to depend upon them."

"The reason for not showing more favor to sprinklered risks, is because our experience with sprinkler systems in this city has shown them to be unsatisfactory, and not to be depended on."

"Stairs should be of steel without any wood sides; if any wood in the construction then there should be sprinklers. Should be sprinklers in all elevators even if they are enclosed, for an elevator is a bad air shaft. Brick factories cut up with wooden partitions are generally hard fires to fight."

"I do not approve of small rooms in factories, they make it very hard for a fireman to fight his way through smoke trying to find a fire when a building of this kind is partitioned off so much."

"In considering the limiting of height and area of a building, the question of accessibility should play an important part."

No. 1393 *d*

# THE PROTECTION OF MAIN BELT DRIVES WITH FIRE RETARDANT PARTITIONS

By C. H. Smith,[1] Boston, Mass.

Non-Member

The importance of safeguarding stairways by placing them in towers well cut off from the remainder of the building and of protecting the openings made by elevators through the floors has long been recognized. Today more than formerly, these features are taken care of in the design of manufacturing buildings, including also well arranged towers for the main belts or ropes where this method of driving is employed. Fig. 1 shows how these features may be taken care of in a textile mill.

2 The following remarks apply more particularly to the older manufacturing buildings and to those of more recent construction where the best principles of design of stair and elevator towers and belt and ropeways have not been followed. Neglect to safeguard vertical openings through floors has resulted in serious loss of life among occupants of the building, who found themselves cut off from their accustomed exits by the rapid spread of fire up through such unprotected openings.

3 In mills insured with the Mutual companies stairs and elevators have generally been well arranged, and the fire protective devices such as automatic sprinkler systems, etc., have shown their value not only in reducing the loss of property by fire to a minimum, but also it has been demonstrated that approved construction, high standards of general order and neatness and efficient fire protection works as well to safeguard the lives of operatives employed.

4 At the present time there are approximately 1,500,000 people employed in the 2800 industrial works insured with the Mutual companies, located in 29 states of the Union and Canada. Since the inception of the system in 1835, there have been but 32 deaths caused directly by fires in these properties and 21 were in a fire in an un-

[1]Engineer and Special Inspector, Associated Factory Mutual Fire Insurance Companies, 31 Milk Street.

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

sprinklered mill in 1876 before sprinklers were in general use. This would indicate that under present conditions, the loss of life would average less than 1 per year per 1,000,000.

5 Of the total of 32 lives lost, poorly constructed beltways which allowed the rapid spread of smoke and flame were to a large extent responsible for the deaths of 25 persons. The need of safeguarding the vertical openings through floors around the main driving belts

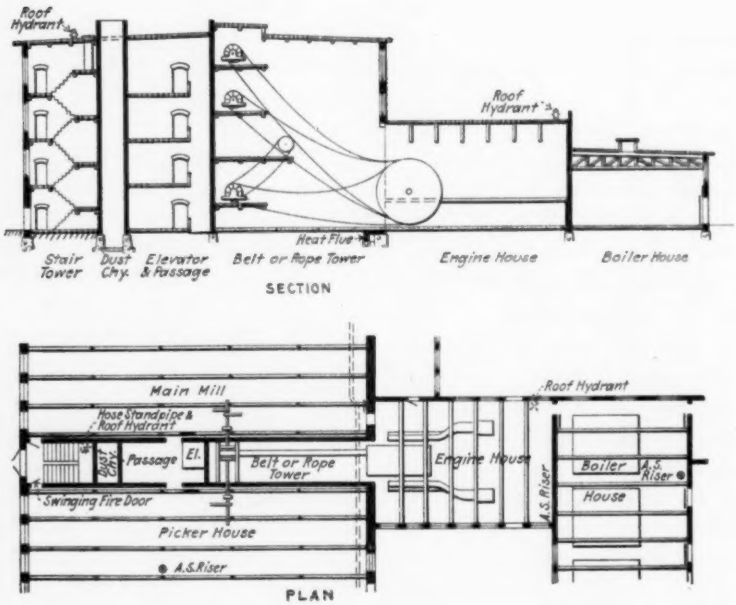



FIG. 1 BELT, STAIRWAY AND ELEVATOR TOWERS

had been less fully appreciated. Conditions at these drives were aggravated moreover, because it was the general custom to enclose the belts with boxes of wood, which in some cases were about head high and in others extended to the ceiling. The boxes tended to become oil soaked and to accumulate lint. A fire once starting at or near them would rapidly make headway, being carried by the natural draft up through the mill. Such a fire would also be more or less sheltered from the action of the sprinklers in the room.

6 The recurrence of several large property losses from this source led to consideration of this matter and measures were taken which

have to a great extent eliminated the open beltway hazard from Mutual risks. In the experience of these companies there have been about 20 fires occurring in the vicinity of main drives in which the open beltway was an important factor in the spread of the fire. These 20 fires resulted in a total loss of $2,721,635, an average of $136,082 per fire. Some of the larger of these losses occurred in the days before sprinkler protection was as complete as now, but the statistics showed that even with complete protection the open beltway was a serious hazard.

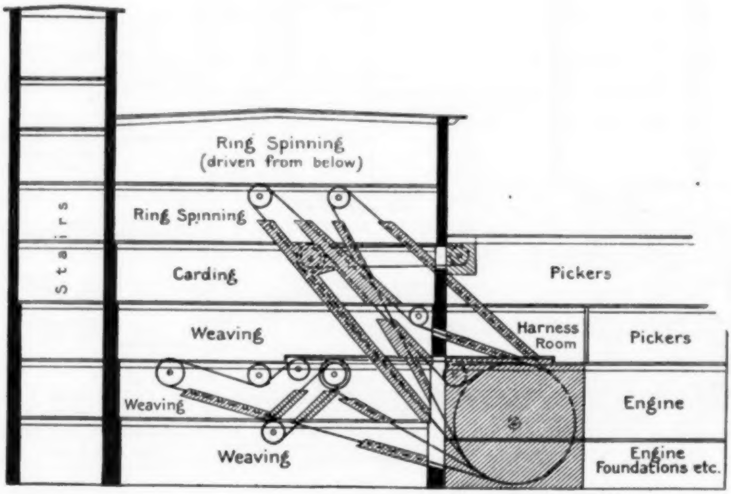



FIG. 2 SECTION SHOWING BELTS AND WOODEN BOXING BEFORE FIRE OF SEPTEMBER 15, 1907

7 The last bad fire from this source occurred September 15, 1907. at a cotton manufacturing establishment in Fall River. This is a stone mill, 339 ft. long, 74 ft. wide and five stories and basement in height with a 4-story wing, 94 ft. long and 65 ft. wide, projecting from the rear at the center of the mill. The engine room was located in the first story of this wing. The belts were boxed with wood and most of these were cut off head high in the several stories. Fig. 2 shows the general arrangement of the drive.

8 Sunday forenoon a bearing in the beltway just above the flywheel was being repaired. While the man doing the work stated that he had no knowledge of anything that could cause the fire, it is

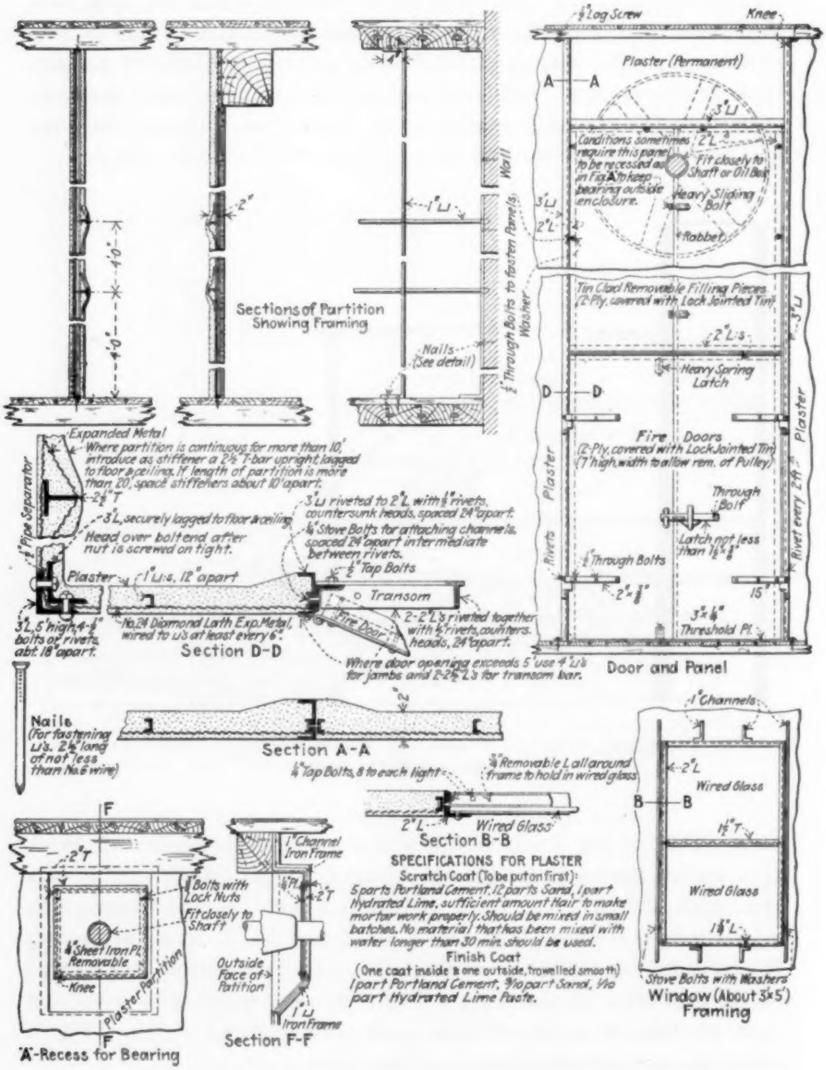



Fig. 3 Details of Construction for Fire Retardant Belt Enclosures

probable that its origin was connected with his work. After completing the job he left the locality. On returning 10 minutes later, he saw fire just below where he had been at work, and gave the alarm.

9 The fire passed up through the wooden belt boxing into all stories as far as the fourth floor where the drive terminated. The mill filled with heat and smoke so rapidly that in 5 minutes no one could enter the rooms. This was in spite of 650 sprinklers which opened, but in justice to the sprinkler equipment, it should be stated that the water pressure at this mill was weak. A section about 50 ft. wide was badly burned on each side of the main drive up through the mill.

10 After this fire plans were worked out to enclose the main drives with partitions of a fire retardant character, so as to approximate the standard belt tower with brick walls, such as are found in many mills of modern design.

11 The limitations of cost, available space, etc., which prevail in many places where the belt tower is not a part of the original design, make necessary special construction such as was adopted in this case, and has been successfully used in many others of the older mills.

12 The plan provided for inclosing the main drives with partitions of expanded metal and cement construction from 2 in. to $2\frac{1}{2}$ in. thick depending on the story heights. A framework is constructed of expanded metal wired to 1 in. or $1\frac{1}{4}$ in. channel iron studs spaced 12 in. apart, and secured to the floor and ceiling. Longitudinal stiffeners of the same material as the studs are used. Where necessary, as in the case of a continuous partition of more than 10 ft., additional stiffness is secured by providing $2\frac{1}{2}$ in. tee-bar uprights. On the frame so constructed portland cement mortar is applied by plastering to make a solid partition, all of the iron frame being embedded in the cement with the exception of the door jambs. These partitions, being comparatively light in weight, could be set up anywhere on the heavy mill floors without the necessity of strengthening them, although where possible it was arranged to have the partitions come over the beams. Although this form of construction for partitions has been largely used and with satisfaction, it would be possible of course to employ some of the special forms of studding now on the market which combine the studs and lathing in one sheet of metal. Details of the construction used are shown in Fig. 3.

13 While in general the enclosures occupy only the floor space necessary for the main belts, it was endeavored to have them as roomy as conditions of machinery installation would permit, in order to facilitate inspection and repairs to the main belts. Provision was made for taking down the lineshafting without disturbing the body of the partitions, usually by placing the fire doors which gave access to the enclosure under the lineshaft, and providing removable wood tin-clad panels constructed like fire doors above the latter. The main bearings were generally left outside the enclosures and to accomplish this the panels in front of the pulleys were sometimes recessed.

14 It was also the endeavor to arrange these enclosures so that they would be as well lighted as possible by including in them windows in the side wall of the building or providing wired glass windows in

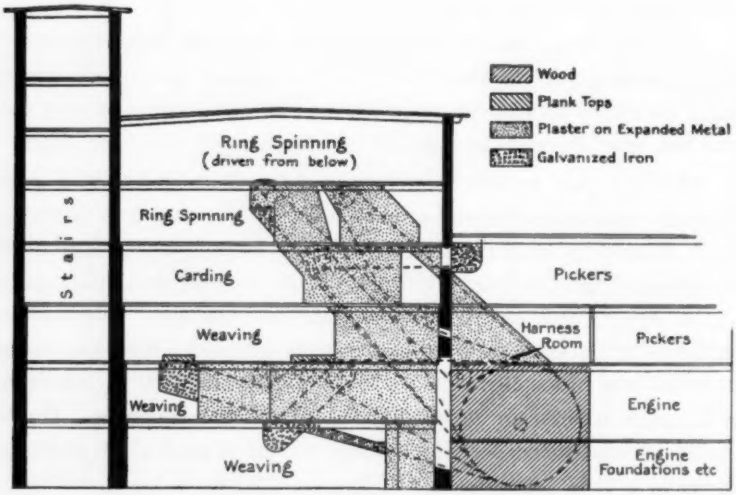



Fig. 4 Section showing Main Drive as now Protected by Fire Retardant Enclosures

metal frames to admit light to the beltway from the room. Fig. 4 shows diagrammatically the completed work at the Fall River mill, and Figs. 5, 6, 7 and 8 are photographs of belt enclosures in different stories. The adaptability of the construction is evidenced in the sloping sides and offsets which it was necessary to make in many cases on account of crowded conditions in the vicinity of the main belts.

15 While there is no claim that these partitions are as efficient

in withstanding the action of a severe fire as a brick wall would be, they are undoubtedly effective in preventing the dangerous draft up through an open beltway. In an actual fire in one of the mills where this construction was installed these enclosures were successful in confining the fire to narrow limits, and undoubtedly prevented a very serious loss.

16 *Stairways.* Where interior stairways are not properly enclosed in brick towers, it is possible to improve the conditions with enclosures of the same type of construction as employed in the beltway work, although it would be much better where the appropriation can be secured to build a standard tower of brick or concrete, especially if the mill is of any considerable height. Placing the stairs and elevators in towers projecting from the mill wall frequently results in a gain of valuable floor space.

17 The type of stair tower that has been developed in the factory buildings at Philadelphia is deserving of more general adoption as it combines with its functions of a stair tower that of a fire escape in the best sense. It consists essentially in a tower separated from the mill so that access to it can be had from the several floors of the mill only from an outside platform or from a vestibule which is open to air. Such a tower can never become filled with smoke from a fire in the mill. Many of the older mills in other sections of the country have stair towers that can be readily converted into towers of the Philadelphia type by closing the openings between the stair tower and the mill in the several stories and arranging for an outside platform in each story communicating from the mill to the tower.

18 *Elevator Enclosures.* We have also found the use of expanded metal and cement partitions practicable for enclosing elevator wells that were not properly protected in the original construction of the building, or where they have since been added. The necessary openings at such elevator shafts should be closed, preferably with wood tin-clad doors of the type which serve as safety gates as well. Where space does not permit of the installation of such doors, rolling steel shutters arranged to be automatically operative by the melting of a fusible link, as well as manually, can be used providing the hazards of occupancy are not excessive.

19 *Other Uses.* The average cost of partitions of the construction advocated is from 30 cents to 33 cents per sq. ft. These figures are for the work in place and include a contractor's profit. These partitions have been used with superior results and not greatly in-

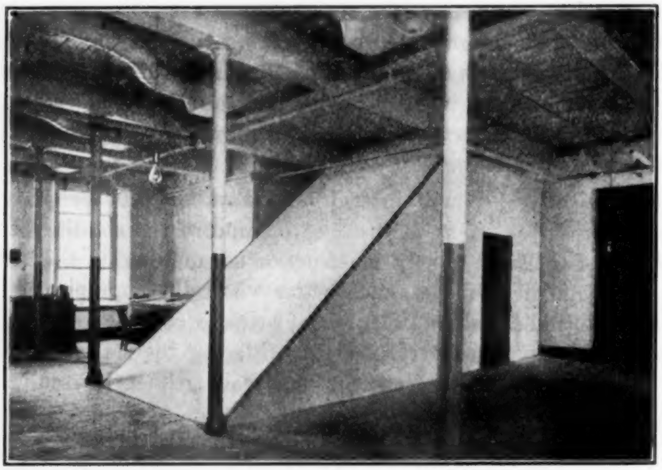


Fig. 5 Harness Room, Second Story, Directly over Flywheel showing Protection of Belts leaving Wheel

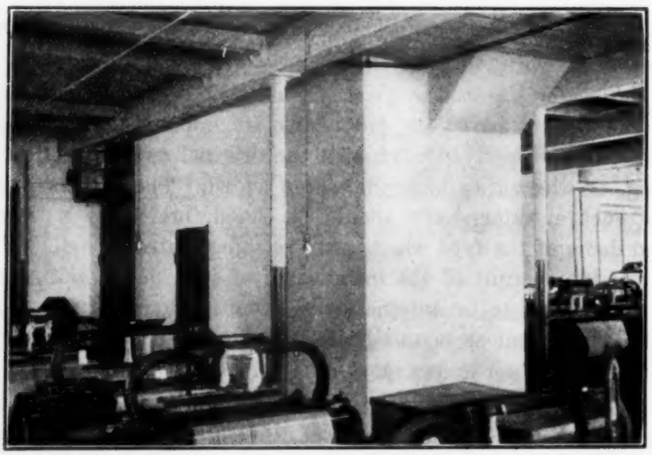


Fig. 6 Weave Room, Second Story. Note Fire Door with Removable Panels above to allow Access to Pulley on Lineshaft

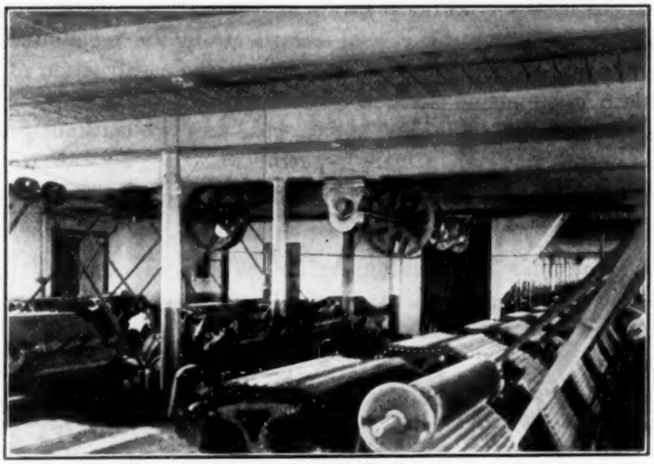


FIG. 7 CARD ROOM, THIRD STORY. ENDS SLOPED TO ECONOMIZE SPACE. NOTE WIRE GLASS WINDOW AND FIRE DOOR WITH REMOVABLE PANELS ABOVE

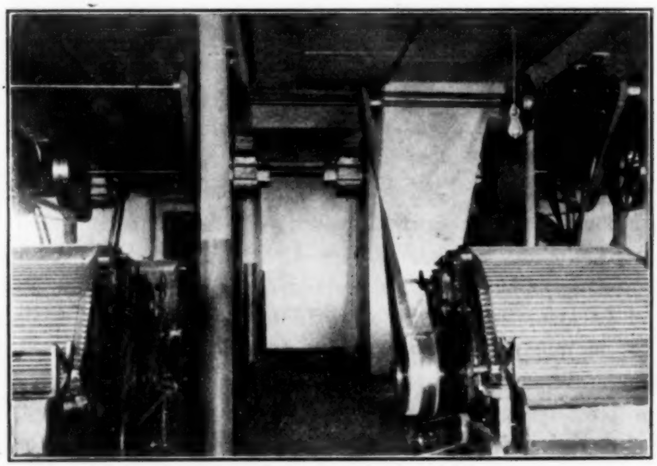


FIG. 8 CARD ROOM, THIRD STORY. END VIEW OF BELT ENCLOSURE. BEARINGS ALL OUTSIDE

creased expense over ordinary forms of combustible construction for the purpose of separating special hazards from the remainder of a manufacturing room. For such purposes as the construction of bins to contain inflammable stock, the segregation of waste working machines, construction of lacquer rooms, etc., uses are constantly being found for this material in manufacturing works.

No. 1393 e

# THE LIFE HAZARD IN CROWDED BUILDINGS DUE TO INADEQUATE EXITS

By H. F. J. Porter, New York

Member of the Society

Buildings in general are either non-fireproof or fireproof. The former can be compared to a pile of kindling wood out in the open, sometimes oil soaked and always ready to be set on fire; the latter to a stove full of fuel ready to be set on fire. In both cases the human occupants swarm around in the interstices in the pile of fuel, and as soon as the fire starts those caught in the fagots have to work their way down through the smoke and flames to the ground to safety.

2 Factory buildings in particular are sources of great danger to their large number of occupants, both on account of their non-fireproof construction and because of the obstructions to rapid egress, due to haphazard placing of machinery, furniture and partitions and the small number, size and character of the exit facilities.

3 Of late, the unrestricted use of fireproof construction in the buildings themselves has been advocated and the author has recommended the development of a form of exit drill of the occupants of each building to determine if, in the case of danger, they could escape readily from the building and if they could not, the alteration of the exits until they could. By "readily" is meant within three minutes, for from many conferences it was found that people do not want, nor would it be safe, to remain in a burning building longer than that time.

4 The capacity of a stairway, if time is not a factor and a stream of people pours into it only at the top and out of it from the bottom, is unlimited; but if time is to be considered the capacity is limited by its cross-sectional area. In a multi-storied building with crowds of people on each floor trying at different points in its length to get on to one stairway in a limited time, the conditions are very different. If more people try to get on to the stairs from each floor than the section between that floor and the floor below will

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

hold, a jam will occur so that the flow downward will cease. The capacity of this section is very limited.

5 A crowd of people does not flow like a liquid composed of round smooth molecules. Their soft bodies are angular in shape more like pieces of rubber with wires in them and they therefore interlock. Clothes present rough surfaces causing friction and if the stairway is narrow an arch is apt to form across it which can become an obstruction in case of pressure from above such as actually to burst the stair rail or enclosing partition.

6 The capacity of a stairway of the average height of from 10 to 12 ft. between floors and not less than 22 in. wide would be one person to every other step or 10 and 12 per floor respectively, and if the width is doubled (not less than 44 in.) so that two people can come down abreast, twice those numbers or 20 and 24. If a stairway has winders in it, its capacity is reduced 50 per cent. One person can descend a single flight of such steps 10 to 12 ft. high in 10 seconds, striking a gait which he can maintain for seven or eight flights of steps. After that he goes slower, making the tenth flight in about 11 or 12 seconds. Every person added in single file adds 1 second to this time. A double file takes no longer if the stairs are double width. Thus it will take 10 seconds for 10 or 20 people, that is, the full capacity of a flight of steps, to come down one story. The capacity of a stairway may be thus increased by widening it in multiples of 22 in. A crowd of people cannot be depended upon to come down more than ten stories. One or more of them will give out, and demand the attention of others. Those who do get down will be severely taxed. The total time required to empty a building is determined by the time required to empty either the floor farthest from the ground or the floor occupied by the greatest number of people.

FORMULA FOR EMPTYING A FLOOR BY ONE STAIRWAY

| | |
|---|---|
| Number of couples (number of people divided by 2) | $c$ |
| Time of formation in line after signal, seconds | 10 |
| Time one couple takes to march to top of stairs, seconds | 10 |
| Time each couple takes to pass through door at top of stairs, seconds | 1 |
| Number of stair flights (one less than number of floors) | $f$ |
| Time of one couple to descend one flight of stairs, seconds | 10 |
| Time of one couple to go from foot of stairs to street, seconds | 10 |

$$\text{Total time} = T = 30 + c1 + f10$$

*Example* Time of emptying 100 people from tenth floor

$T = 30 + 50 + 90 = 170$ seconds $= 2$ minutes, 50 seconds

*Example* Time of emptying a ten-story building with 20 people on each floor is the same as emptying 20 people from tenth floor.

$$T = 30 + 10 + 90 = 130 = 2 \text{ minutes, } 10 \text{ seconds}$$

7 Tests of the capacity of fire escapes in a limited time gave the following results: A straight ladder, 2 per floor; ladder set at 50 to 60 deg. with the horizontal requiring people to go down backwards 3 to 4 per floor; stairs 30 in. wide, 10 to 12 per floor; and the modern outside stairway with a mezzanine platform 40 in. wide, 20 to 24 per floor, the same as an inside stairway. Fire escapes are usually so exposed to flames from windows opening upon them that they are more often fire traps than fire escapes. They should be prohibited by law and safer methods of escape provided.

8 In order to insure the safety of the occupants of a building in case of emergency one of two things has to be done: (*a*) there should be two stairways so that if one is cut off by flames or smoke the other can be used and the number of occupants reduced on each floor to meet the limited capacity of the part of the stairway between floors, or (*b*) the number of stairways increased so as to have two separate and independent stairways from each floor to the ground with its own exit from the building. People can then pour into the top of whichever one is not cut off by the fire and continue down and out at the bottom without colliding with those from any other floor. Fire drills installed under either of these conditions worked more or less satisfactorily, and the author tried unsuccessfully for years to have ordinances passed in New York City and legislation enacted at Albany, making them mandatory, but the expense of changes in the buildings and the idea of having employees walk out of a factory while manufacturing operations were under way, upon the sounding of an unexpected signal, did not appeal to factory proprietors as practical. It required holocausts in New Jersey, Pennsylvania and New York finally to bring about the legislation in those states.

9 As time passed, however, the author developed what might be termed an exit test in factories which presented the opportunity and found to his astonishment that almost without exception, exit facilities adequate for handling the regular number of occupants under emergency conditions, were lacking.

10 This situation has probably developed with the rapid growth of industry where a factory building had been built to accommodate a certain number of people, and then, as the business grew, more people were accommodated without realizing that each additional person became an increment of danger to all. Or, if the danger was

at all appreciated, some means of escape from windows was supplied which might be anything from a rope to a ladder. After this condition had become general it crystallized into custom, and new buildings with exit facilities inadequate for their occupancy were designed, erected and accepted as safe. Ropes were followed by ladders, and these in turn by fire escapes which became in time an established necessity.

11 Engineers, when called upon to supply a mechanism, are expected to have it subjected to a working test, which it must pass before they get paid for it; but architects and builders have never been called upon to demonstrate by actual test that the facilities which they have supplied in their buildings for the purpose of emptying them under emergency conditions will actually work, and this notwithstanding repeated instances of panic congestion on stairs, of people being burned to death on fire escapes, of elevators sticking from the warping of their runways from heat, etc.

12 When subjected to test these exit facilities in many buildings have been found to be entirely wanting in adequacy, and when this fact was brought to the attention of those who were responsible, it has been surprising to find how readily they accepted the criticism. On the other hand, those who possess these unemptiable buildings are sceptical of such statements and unwilling to be persuaded that the buildings are not safe. They point to all the other buildings erected by reputable architects and builders and naturally are incredulous.

13 In order to empty these buildings, additional stairways had to be built and fire drills developed to take the people out. Such changes in the building are expensive, for two stairways have to be installed from each floor to the ground, so that if one is cut off by a fire, the other can be used. In many-storied buildings the number of stairways required becomes impractical. In addition fire drills are expensive to operate, for they involve not only loss of time of operatives and a break in the continuity of the process of manufacture, but the actual going down stairs and return of people, some of whom may be lame, others affected by a weak heart or lungs, others anaemic or organically weak, reduce the efficiency of the working force for a very appreciable time. If the drill takes place at the end of the day this criticism might be modified slightly.

14 Such is the situation in the usual type of factory building to be found in the average town where ground is cheap, buildings large and stairways broad. Turning now to the loft building used for

factory purposes, the conditions as regards emptiability are found to be very much worse and have to be corrected in a different manner.

15 Let us consider for the moment a one-story or ground-floor factory building with a doorway at each side, one of which is cut off by a fire. The people can march out horizontally through the other doorway and nothing will inpede this horizontal exit except the size of the doorway. If this is 22 in. wide, a single file of people can pass out in an orderly manner at the rate of one person every second. If

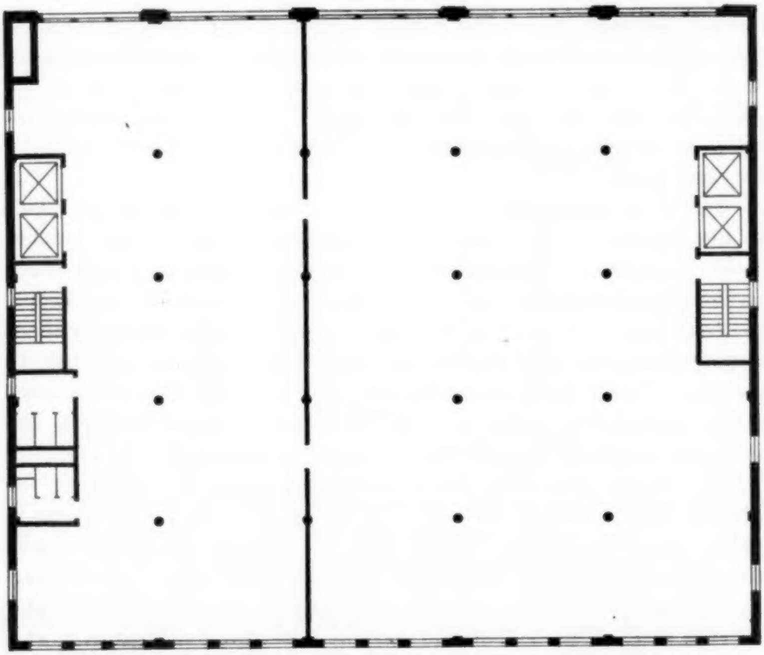


Fig. 1 Floor Plan of Typical Loft Building showing Fire Wall with Doorways

it is 44 in. wide, a line of people two abreast can pass out in the same time. One hundred people can make their exit through one 44-in. door, therefore, in 50 seconds, or say one minute.

16 Now put another factory on top of this one with one hundred people in it. The doorway at each side will have to open on stairways which lead down to the doorways constituting the exits from the factory below. Suppose a fire occurs on the floor below, cutting off one of these exits, the 100 people on the lower floor immediately

proceed to make their horizontal exit, while those on the upper floor proceed to make a vertical downward exit to reach the doorway out of which those below are moving. The result is of course a collision, the stream of people from upstairs coming down upon the stream of people on the ground floor on their way out. This collision prevents both the upstairs stream from coming down and the down-stairs stream from going out. There is a complete lock, and the building does not empty.

17 Not only have we put one factory on another in the case of our loft building, but we have piled factory on factory until we have from 10 to 30 and more, one on top of the other; and each employing from 100 to 300 or more people. In cases of emergency as in the Asch Building fire, there are only two courses for the occupants: one is to burn to death, and the other to jump to death—"to burn up or jump down."

18 It is impossible to reduce the number of people per floor to the capacity of the stairs, say 24 per floor. Even if that number would be all that a business required, in case of emergency they would have to go downstairs, and it is a physical impossibility for people to stand the exertion of a trip down more than ten stories without resting; and when they stop to rest they block the stream and obstruct its exit. Under these circumstances it is necessary to develop some other method for people in high buildings to secure safety. The following suggestion is offered to meet the situation:

19 It has been seen that a horizontal escape by people on the ground floor is readily secured. Let us see if a horizontal escape to safety for people at any height from the ground can be developed. Suppose a wall is built across the building from cellar to roof practically bisecting it in a way so as to have a stairway and elevator on each side. This wall should have at least two doorways in it at a considerable distance from each other and closed by self-closing fireproof doors (Fig. 1).

20 It is improbable that a fire will occur on both sides of this wall simultaneously. It could occur only by incendiary origin, and that would hardly be possible in working hours. Should one occur on either side, the people on that side would go through the doorways in the fire wall, close the doors after them and be perfectly safe. That half of the building in which the fire might be should be emptied in less than a minute if there were no more than 100 people on each floor to pass through one doorway 44 in. wide. If the principle of the horizontal escape presented by the fire wall is included

in the design of new buildings a most satisfactory method of securing safety at comparatively small expense will be offered.

21 In every way possible the horizontal escape should be developed in old buildings and the vertical escape subordinated. Factory buildings adjoining one another may have doorways through their sides connecting them on various floors closed by fireproof self-closing doors, or may be connected by outside balconies built around the party walls; or, if of different heights, doors in the sides of one may lead out on the roofs of the others.

22 The fire wall bisecting the building as described makes practically two buildings, each provided with elevators and stairways.

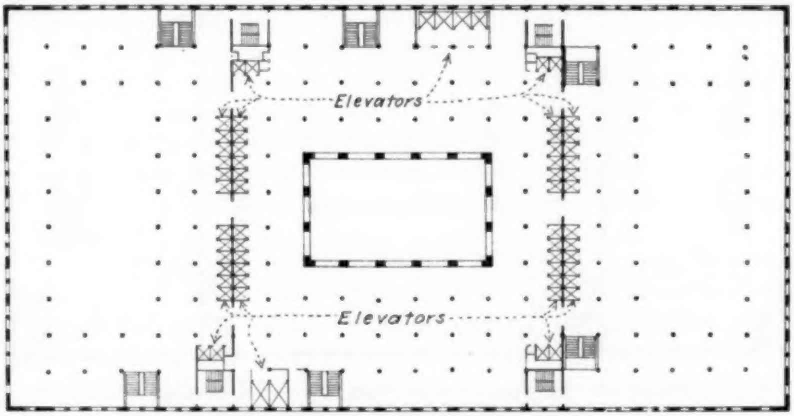



Fig. 2 Department Store Floor Plan Showing Present Arrangement of Fire Walls, Elevators and Stairs

A fire on one side of the wall would be confined to half the building, and therefore the property loss would be reduced one-half. Only one-half the people would be endangered and have to move, and the distance they would have to go would be only one-half what it would be if they were on the ground floor of a building without a fire wall. They could remain on the same floor till the fire was extinguished, or could go down to the ground by the elevators operating under normal conditions.

23 The fire wall eliminates the necessity for a fire drill with its accompanying objections. Of course all buildings occupied by many people should have a fire alarm signal system in them to advise the people promptly of their danger. In buildings where there is a fire

wall the signals should be arranged so that in case a fire should occur on one side of the fire wall on any floor, a bell on each floor on the same side of the fire wall would ring, indicating on which floor the fire is. Then all the people on that floor and above it should pass through the fire wall and close the doors. Those below need not disturb themselves until the fire threatens them, and then they too can pass through the fire wall.

24 There are certain other safety devices which should be supplied in factories to protect the lives of the operatives from fire.

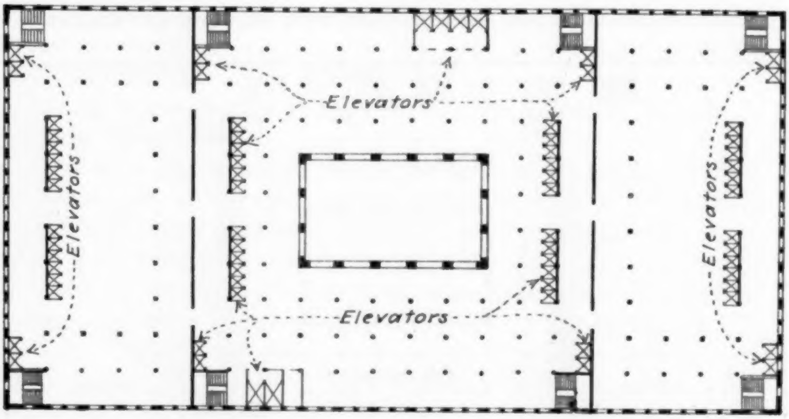



FIG. 3 SUGGESTED ARRANGEMENT OF FIRE WALLS, ELEVATORS AND STAIRS FOR DEPARTMENT STORE

One of these is metal-framed windows with wire glass. These are made so as to close automatically in case of fire, thus preventing the latter from spreading upwards from floor to floor outside of the building.

25 Another safety device is automatic sprinklers which serve to extinguish fires in their incipiency. All doors should be made to swing outward, and where they open on a hall or stair landing they should be vestibuled, so as not to obstruct the passage way. Sliding doors should be avoided if possible, as they are apt to stick or jam by pressure of people upon them.

26 Each floor of our typical loft buildings is say 100 ft. by 100 ft. by 10 ft. and therefore contains 100,000 cu. ft. of air. The laws of New York and many other states require 250 cu. ft. of air per person as a limitation of occupancy. This limits the number of

people per floor in a building of this size to 400 and if the stairways were 44 in. wide (and there are none now over 36 in.) at most only 40 per floor could possibly go down them even if the other 360 would let them.

27 With the fire wall only 200 of the 400 people on each floor would have to move, and if there were two doorways in the fire wall at some distance from each other, they could reach safety through them in one minute, or if one were cut off by the fire, all could pass through the other easily in two minutes. More doorways can be introduced, and thus the time of exit could be lowered still further.

28 An effort is being made to increase the amount of air space required per person from 250 to 500 cu. ft., which would reduce the number of people per floor to 200, of whom only 100 would have to move, and they could easily reach safety in one minute.

29 The stairways and elevators should be enclosed in fireproof walls to prevent a fire on one floor continuing upward and setting the other floors on fire. The ceiling of the basement where the machinery is located should be fireproof, and should not be pierced inside of the building, so that a fire there would not reach the elevator shafts.

30 Fire escapes which are simply stairs and possess dangerous features not only of limitations as to size, but of accessibility for flames and smoke, should be looked upon as evidence of the incompetence or ignorance, or worse, of the architect, builder, or owner, and prohibited by law under a heavy fine. They are not only dangerous to life by giving a false confidence in their adequacy for escape, but they destroy the appearance of the building. Our cities should be built without such architectural blemishes.

31 Fire escapes of the chute type are tubes with a smooth helix instead of steps. If the only opening is at the top they have considerable capacity. They soon rust, however, and at best are not to be considered seriously in comparison with other means of safe exit. People cannot enter them at different floors while a stream of people is passing down from above.

32 The smoke-proof tower, claimed to have originated in Philadelphia, is the latest improvement in the line of fire escapes. It is simply an enclosed stairway on the outside of a building, but cannot be reached except by going out of doors. Its special claim is that smoke and flames cannot get into it. It has, however, no more capacity than any other stairway, and as its approach is always open to the weather and its interior is always more or less

dark, it is never used in ordinary service and becomes neglected. These monuments to architectural incompetency can be seen here and there filled with the dust and accumulated rubbish of every unused open space. When a time arrives for using them everybody has forgotten their existence. During the last year or two, notwithstanding the protests of many, a great many new buildings have been constructed, especially in New York City, with these monstrosities on them, and have been accepted by the building department in all seriousness.

33 The fire wall should be introduced into all buildings where the public congregates in large numbers. Large department stores, which on certain days are said to accommodate several thousand people per floor, are very dangerous places at present. A fire, or a panic without a fire, might cause a fearful tragedy. It is criminal for their owners to object to fire walls and offer as an excuse that they would obstruct the vista. Certain cities require fire walls in such buildings now as a property protection, and the vista is dispensed with without comment. The department stores of Philadelphia are so divided; John Wanamaker's new store there is divided by two such walls as shown in Fig. 2. The exit facilities in it, however, are badly arranged, for the architect apparently did not think of the life hazard of its occupants, and designed the fire walls to protect property only. Fig. 3 shows how the building might be redesigned so as to be safer. It should be noted that the elevators are removed from the fire wall so that people trying to go down in them would not block the doorways of the fire wall and prevent others coming through them. The stairways are situated as far from the fire wall as possible and should be enclosed by fireproof partitions.

34 Churches, assembly halls and similar ground-floor buildings should have their floor fireproof and unpierced so that any fire occurring in the basement would not endanger the occupants of the main building.

35 Moving picture buildings, theaters, etc., should be redesigned (Fig. 4). People come out of them by the way they go in, and in case of emergency all crowd into the narrow aisles. These aisles should be turned across the room and lead directly to courts opening on the street in a way such that streams of people will not collide. The various balconies and galleries should have foyers behind fire walls with separate stairs and street exits so that patrons will not have to mingle with those making their exit from the lower floors.

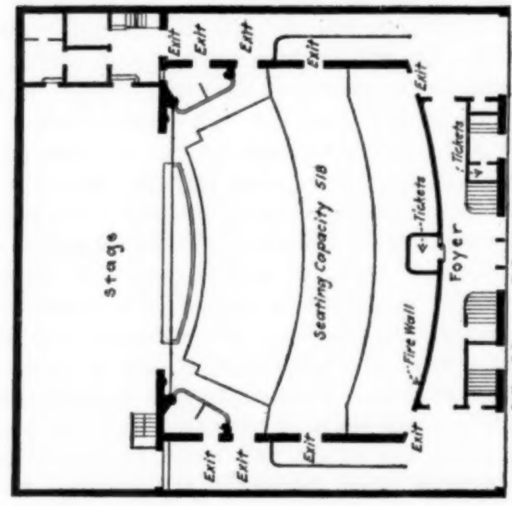



FIG. 4 TYPICAL AND PROPOSED THEATER PLAN SHOWING USE OF FIRE WALLS, SIDE ENTRANCES AND EXITS

36 Every school building should be divided by a fire wall providing a horizontal exit on each floor, so that the children will not have to be drilled to go downstairs in case of fire.

37 Hospitals where the inmates are bedridden, blind, lame, invalid, imbecile, or otherwise helpless, can be made safe by the introduction of the fire wall between wards, and in case of fire those who are bedridden can be wheeled on their beds through the doorways, and those who are up and about can walk through them.

38 Hotels and apartment buildings can so easily have a fire wall developed in them that it need only be referred to here in passing. Even the private residence where only a few pople occupy a floor can be made safe in this way. The back stairway should be enclosed in a fireproof partition, and in case of a fire instead of everybody having to go downstairs through the smoke and flames, or having to jump from windows, the people on each floor have simply to pass through the fireproof door and go down stairs in safety. In large residences where there is a servants' quarters in connection with the back stairs, the building would be bisected and the people on either side of the wall would be able to carry their clothing and perhaps much household and personal property to safety.

39 Two years ago the National Fire Protection Association appointed a committee of which the author was a member to draft suggestions for the organization and execution of fire drills. This committee made its report to the annual meeting of the association held in Chicago last May, and it was adopted with slight modifications. A prefatory note to this report is as follows:

> Many so-called fire drills, outside fire escapes, and similar practices and devices are generally insufficient, often dangerous, and therefore misleading substitutes for rational exit facilities, and are a manifestation of improper design and construction of our buildings. A stairway connecting many stories will accommodate only a limited number of people. Stairways are, therefore, dangerous means of exit for crowds. Congestion is bound to occur in them when used under stress of excitement owing to their limitations.
>
> The primary object of the exit drill is to determine if the building is properly designed so that in the emergency of a fire its occupants would be able to effect their escape readily without the probability of injury from stairway or other congestion which inevitably causes panic. This test should be occasionally repeated to insure the continuous maintenance of safe conditions.

40 The author advocates legislation, requiring three things: (*a*) Architects and builders should be prohibited from designing buildings which cannot be emptied within 3 minutes after a given

signal. (*b*) The municipal authorities should be required to institute an exit test in each building to determine, before it is accepted, if it can be emptied of its occupants in 3 minutes. If it cannot pass this test it will not be accepted and must be altered until it can pass the test. (*c*) Afterwards the proper authorities will be required to repeat the exit test from time to time, to see that the safe conditions originally established are maintained.

## No. 1393f

# DISCUSSION ON FIRE PROTECTION

W. H. Kenerson referred to the statement of Mr. Porter that people do not want, nor is it safe for them to remain in a burning building more than three minutes. He said that he was present at the start of a very quick fire where the people in the building were reluctant to stay at all. They got out of the building by jumping almost as quickly as they could run to the side walls. Even with adequate exits it was plainly evident that danger could not be eliminated. In buildings where operatives were working under crowded conditions some people would be burned or crushed before they could get out in case of a severe fire, even if the walls were open all the way around, owing to the furniture, machines, etc., being closely grouped. The panic following an incipient fire was often worse than the fire itself and did more damage. In some of our large cities, the streets in the neighborhood of office buildings and factories are so narrow and so hemmed in by the crowds rushing toward the building in case of a fire that there would not be room enough in the street to accommodate the operatives who were leaving. This was certainly true of some loft buildings in New York City.

Regarding the use of the fire wall, while it is of great value in certain cases and places, it is not altogether dependable. Was it not conceivable that fires could not start on both sides of a partition at once? There were many conditions under which a fire would rapidly spread to both sides of a partition even if it did not start on both sides, as for example in a flash fire.

The author very properly said that sliding doors inside of partitions were dangerous; but what of swing doors? These should always open outward, but what was "outward" in a partition that must be used in both directions? Where communication was established between adjacent buildings even, it was inevitable that one of the doors would open the wrong way.

There was only one paragraph in the paper about preventive methods through the use of automatic apparatus in conjunction with other things for preventing fire at the start. It was unfortunate that this was not included in a paper of the general scope of the one under discussion.

---

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

He believed that it would be impossible to enforce the legislation recommended in the last paragraph of the paper to prevent architects from designing buildings which could be emptied in 3 minutes. Before construction such a matter was a question of opinion rather than fact. Moreover, operatives could not be prevented from crowding or becoming panic stricken in the event of a fire and it was doubtful under such circumstances whether many a properly designed building could be emptied in 3 minutes even if it were possible to enforce the first recommendation.

Henry Hess. I am neither an insurance nor a sprinkler man. I have had some experience putting up factory buildings and running the establishment afterwards. One of my most interesting experiences of that character was in building a large tool works in Germany. The first thing I ran afoul of was the law. I bring this up particularly because Professor Kenerson has indicated his belief that adequate legislation was absolutely impossible, if not as to securing it in the first place, then as to its later enforcement. Now that is a question of the will of the community. If the community really desires enforcement of law it will see to such enforcement. Fire wall partitions such as Mr. Porter advocates are very useful. In the case of the German factory partitions were installed in certain cases and were not insisted upon in others; but there was an insistence upon one other thing of possibly far greater importance. The authorities determined a central point for each floor area, considered the number of people at work in that area, and then insisted that there must be an adequate fireproof staircase within a certain distance of that central point. The size of the stairway was made to increase with the increase of that distance, which latter was not permitted to exceed a certain limit. Moreover, a stairway had to be provided not merely at one end, but at two ends at diagonal corners so that there was a possibility of reaching a stairway in two directions from a central point. The law does not permit overcrowding. Such conditions as obtain in New York City loft buildings would not be tolerated in the countries from which the people who run the industries in these lofts have come to us. Why should our communities grant them such permission here? Panic cannot be altogether guarded against in the construction of a building. Sprinklers do not constitute a full safeguard. In fact if you were to send down a douche of water upon an excitable woman when she smells smoke or sees fire you would most certainly not cool her down. Provide what safeguards you will, always keep

the crowds down to a point where even panic stricken people cannot seriously jam.

Geo. I. Rockwood. As one interested in the general subject of fire protection, I have followed closely the fire statistics of the last seven or eight years as given out by the National Fire Protection Association and the Factory Mutual Fire Insurance Companies. I have also had experience with the manner in which automatic sprinklers operate to save human life, having been responsible for the installation of many hundreds of thousands of sprinkler heads; so that with that experience back of me I want to say that no one so far as I know has really done more than has Mr. Porter to convert the rather obstinate architectural profession, as well as the equally obstinate members of the profession of fire protection engineering, to the use of this very obvious device for getting operatives and others out of a burning room and into a place of safety.

The way of the reformer is proverbially difficult, but it seems to me that those responsible for the design of loft buildings have been particularly slow in appreciating Mr. Porter's work. Very likely the reason for this will be found in Mr. Porter's scarcely concealed contempt for the automatic sprinkler as a life-saving device, but neither Mr. Porter nor anyone else has yet made any proper inquiry into the history of automatic sprinklers, considered solely from the point of view of their use as life savers. They have been regarded altogether as property savers, and are installed in buildings for that purpose. It is perfectly clear that every time a sprinkler operates successfully in a building containing human beings it is potentially a life saver. The question narrows down, therefore, to the effect of sprinklers in preventing panic conditions in buildings. Do they prevent panics or can they be made to prevent them? We need a little further light on this subject before we can dogmatize to advantage either way, but I feel confident that the solution of the problem of saving human life is a function equally of the use of Mr. Porter's centrally located incombustible partitions and the proper use of automatic sprinklers.

In the average case of a fire extinguished by automatic sprinklers three or four heads are all that open to effect the result. In the case of quick flash fires an entire roomful of sprinklers may open, seemingly at once, without any appreciable interval of time between the setting of the fire and the operation of the sprinklers. The question naturally arises, would such instant out-pouring of water be an advantage or a disadvantage to a room more or less

crowded with operatives No sane man would take the ground that he would rather be in such a room, under those conditions, without sprinklers. In a recent case where a flash fire occurred in the first floor of a three-story factory filled with people on every floor, a can of material in which gasolene was an ingredient took fire, with the curious result that, although the room was not a particularly large one, and there were many men at work in it, not a man got out of the room without getting wet. They said afterwards that there seemed to be no interval of time between the instant when the can flashed up and the subsequent out-pouring of water from all of the sprinkler heads in the room.

It might be thought that where a factory has a good deal of material out of which, say, straw hats, or lingerie, are made, the rapidity with which the fire, once started, would propagate would insure the injury of everybody in the room. Such a view would fail entirely to take into account the fact that when the fire starts the heated gases almost instantly rise to the ceiling where they proceed to mushroom out and open the sprinkler heads one after another much faster than the fire can propagate itself in the material on the floor. This is not a matter of speculation; it is an observed fact.

There are classes of buildings in which there are so many concealed spaces in the wooden walls, wooden partitions and wooden floors, that a heavy fire may conceivably get strong headway in some concealed space before it shows itself. Under such circumstances the sprinklers operate at a disadvantage and merely act as a check to the spread of the flames until the arrival of the fire department. But even then they often make it possible for the firemen to carry their hose to the very center of the fire, and thus are the primary cause of its final extinguishment.

Mr. Porter would do well to add to his crusade in favor of a central figure resistant partition, with separate means of egress on opposite sides of it, a further demand for the installation of automatic sprinklers, and he will be surprised to see how much more quickly his original effort at reform will succeed.

Harrington Emerson. In making comparative investigations of American and foreign cities what strikes one most is the much lower fire loss in German cities than in American cities. The reason is that different ideals have been pursued. Our ideal is to get there immediately. In an international test of different fire companies at Berlin about 20 years ago it took the Americans 20 seconds to come out,

couple up the hose and begin to play the water while it took the Germans over 8 minutes to prepare to fight the fire. Nevertheless the fact remains that the fire loss in Germany is small compared with what it is in this country. It is not because they have buildings that cannot be burned, because recently in Hamburg a number of incendiary fires occurred with severe losses. But the principle of the Germans is to prevent fire, not to fight it after it is started; while we have gone the limit in fighting fire after it is started. We are away behind the rest of the world in preventing fires before they start. Switzerland is a country of wooden houses yet the per capita fire loss there last year was only 2 per cent of the American per capita loss.

F. B. Gilbreth. There is no doubt that standardization more than any other one thing will reduce the number of fires; but let it be standardization for the prevention of fires rather than for extinguishing them after they are started. Fires are the product of ignorance. A very small proportion of architects, and a still smaller number of engineers, know how to construct a building that will not burn up.

In a paper on the Waste of Natural Resources by Fire,[1] Charles Whiting Baker gave a striking illustration of the annual loss not only of property but of human life. He said:

The buildings consumed, if placed on lots of 65 ft. frontage, would line both sides of a street extending from New York to Chicago. A person journeying along this street of desolation would pass in every thousand feet a ruin from which an injured person was taken. At every three-quarters of a mile in this journey he would encounter the charred remains of a human being who had been burned to death.

I have seen many of the big conflagrations in this country: Toronto, Rochester, Baltimore, San Francisco and Sioux City. A careful examination of all these ruins shows that if we tried to construct buildings that would burn up we really could not do a much better job. Reports of experiments carried on by Professor Woolson in New York City during the last ten years contain much valuable information upon this subject.

We face an entirely new situation today with the advent of concrete, which when made with the proper aggregates, such as trap rock and the right kind of sand, is of tremendous assistance in resisting the spread of fires. In fact, we are not depending on any of the old

[1]Joint meeting of the Engineering Societies on the Conservation of Natural Resources, New York, March 24, 1909.

types of construction and fireproofing schemes for preventing the spread of fires. Let us have buildings, to begin with, that will not aid the spread of fire.

I suggest for the consideration of the Society an exhibit room to which engineers, architects and insurance companies may come and satisfy themselves that there is not a single thing in construction today that cannot be made better and quite as cheaply out of absolutely incombustible material.

JAMES B. SCOTT. Education, it has been said, should properly begin with one's grandparents, and undoubtedly the time to begin fighting a fire is before it begins. But granting that the "eugenics" of fire fighting have been properly observed, there will always remain the possibility of accident or incendiarism, giving rise to a hand-to-hand conflict with man's ancient enemy. The ships that can deliver more shells and heavier shells, in a given time and within a given area, and can begin delivery a few minutes earlier than the enemy, are usually awarded the victory in a modern naval engagement. The fire-fighting system which can deliver enough water, at a suitable pressure, just where it is needed, and can begin delivery in the fewest possible minutes after the discovery of a dangerous fire, is approaching the coveted 100 per cent efficiency in its limited field, as distinguished from the wider province of fire prevention.

## No. 1394

# STANDARD SIZES OF CATALOGUES

### REPORT OF COMMITTEE ON STANDARDIZATION OF CATALOGUES

The Committee has made an extensive investigation of the various sizes of catalogues in common use, and has had correspondence with many paper manufacturers, printers, advertising agents and makers of filing boxes and cabinets, as well as with manufacturers of machinery and supplies. The general conclusions it has reached may be summarized as follows:

*a* There is a universal opinion that catalogue sizes should be standardized. It is not necessary to give any reasons here for this opinion, as it is all on one side.

*b* The standard of 6 in. by 9 in. has for so many years met with such wide acceptance, probably two-thirds of all the catalogues that are now made being either that size, trimmed as closely to exact size as possible, or within ¼ in. of it in one or both dimensions, that it may be considered as too well established ever to be abolished.

*c* The 6-in. by 9-in. size is too small for many purposes, and it is necessary to have for some purposes a size which is about 8 in. by 10½ in. or 8½ in. by 11 in.

There are objections to having both considered as standard sizes, and as there is an increasing tendency to the use of the larger size the Committee recommends that 8½ in. by 11 in. be the standard. Electrotypes 4 in. by 7 in., which are commonly used for 6-in. by 9-in. pages, may conveniently be used, two on a page, on an 8½-in. by 11-in. page, but they are too large for the 8-in. by 10-½ in. page.

The 8½-in. by 11-in. page folded twice makes 3⅞ in. by 8½ in., which is a convenient size for a long folder.

*d* The 9-in. by 12-in. size has been adopted by a few for very large catalogues. While it is generally acceptable for technical and trade journals, it is rather too large for a catalogue, unless it is to be a large cloth-bound book, to be placed on a shelf and not filed in a cabinet. It should,

---

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers, and later revised.

therefore, not be recommended as a standard for common use.

The following extracts from some of the letters received show the variety of opinions that exists as regards the sizes that should be adopted:

*Matthews-Northrup Works, Buffalo, N. Y.* Paper stock is made in sheet sizes 25 in. by 38 in. Since cuts have been made and tabular pages set for years to fit 6 in. by 9 in. it should prove very expensive for many firms to change. An opportunity is afforded by the 9 in. by 12 in. size to run the text descriptive matter or tabular panel of sizes and prices immediately under the illustrations on the same page. The 8½ in. by 11 in. size works out in good proportions. These three sizes seem to be the only ones that need be planned for.

*Robert L. Stillman, New York.* I suppose there are more catalogues made 6⅛ in. by 9¼ in. or 9⅜ in. by 12¼ in. than any other size, because they both cut from 25 in. by 38 in. paper which is of standard size, allowing ⅛ in. for trim.

The 33 in. by 46 in. size would make 8⅛ in. by 11¼ in. It might be reduced to 11 in. to get it within the standard letter size.

*Ticonderoga Pulp & Paper Company, New York.* It would be much to the advantage of every one if all catalogues could be standardized to a certain size. It would be a great assistance not only to the paper manufacturer but also to the dealer and printer, as in that case the dealer could carry in stock standard sizes and could supply paper promptly, and printers would not then have to wait for paper to be made especially.

*Frank Presbrey Company, New York.* Sent samples 6¾ in. by 9¾ in., and recommended it as the size used in all standard magazines. In reference to letter size, 8½ in. by 11 in., if a movement be inaugurated which will result in making up to this size all catalogues and monthly bulletins a great deal will be accomplished.

*Collin Armstrong Advertising Company, New York.* As far as our experience goes 5½ in. by 8 in. is the most convenient size. Anything near 8½ in. by 11 in. would in our opinion be too large to be handled easily, would be apt to go to pieces quickly and probably not be kept carefully by the recipient, as would be the case with a smaller book.

*F. F. Coleman, Publicity Manager, Lidgerwood Manufacturing Company, New York.* All publications except catalogues intended for filing should be eliminated from consideration. The best size for catalogues is 9 in. by 12 in. upright. Personally, I would abolish the 6 in. by 9 in. size.

*J. Horace McFarland, Harrisburg, Pa.* In the nursery trade the 6 in. by 9 in. size does not permit such attractive illustrations in an economical manner as the growing vigor of selling campaigns requires, and there is a great disposition to get away from anything of a standard size and to make the catalogue different.

*Hammacher, Schlemmer & Company, New York,* are working on a new edition of their catalogue which will be 9 in. by 12 in. For a number of years they used 5½ in. by 7¾ in., but it is too small for a book of say 1200 pages. For pamphlets and smaller sizes will use 6 in. by 9 in.

*Manning, Maxwell & Moore, New York.* Books for general mechanical goods, both small and large should be 6 in. by 9 in. wherever it is practicable. In many cases a larger sheet is necessary to take the proper size cut to show important small details.

*General Electric Company, Schenectady, N. Y.* We use and recommend 8 in. by 10½ in. catalogues as standard. The width of page is valuable for displaying illustrations and tabular matter. Our standard letter paper is 8 in. by 10½ in. Our standard photographic plate is 8 in. by 10 in.

*The Yale & Towne Manufacturing Company, New York. C. L. Redfield, Advertising Manager.* Printed matter may well be any of the following sizes for reasons given: 6 in. by 3⅝ in. for folders, because this size just fits the small Government envelope which is still very largely in use among business people; 8 in. by 3½ in. for folders and small booklets, because it just fits the No. 9 Government 2-cent envelope and makes it very easy to use such printed matter to accompany correspondence; 6 in. by 9 in. for small catalogues, as we have been led to believe that this is more acceptable to the machinery and other trades; 8 in. by 10½ in. for sheets which accompany correspondence, as this is the size of our letter head; 9 in. by 12 in. for our more important catalogues, this being the accepted size with the paper makers and is carried in stock.

From a *Circular of Advice* issued by the American Institute of Architects, April 1913, manufacturers are earnestly urged to adopt a standard size of 8½ in. by 11 in. for all catalogue matter which they desire an architect to keep for reference, this being the size which can be most readily filed in the standard letter files of the day. For convenience in indexing, it is essential that all catalogue matter should be divided into sections, so that the sections of different manufacturers treating on the same subject may be grouped together. This means that only one article should be treated on a single leaf, and that no leaves should be bound together except where they refer to a single article or variations of that article.

A list of 927 catalogues measured for the Technical Publicity Association showed that there were no less than 147 different sizes. Dividing the 147 sizes into eight groups there are the following:

3 to 3¾ by 5 to 9, sixty, of which 34 at 3½ by 6.

4 to 4¾ by 5¾ to 9¼, one hundred and twenty, 21 of which are 4 by 6 and 26 are 4 by 7½.

5 to 5¾ by 6½ to 9¼, one hundred and eighteen, 15 of which are 5 by 7 and 17 are 5 by 7½.

6 to 6¾ by 7 to 10½, four hundred and forty-three, 325 of which are 6 by 9 and 23 are 6½ by 9½.

7 to 7¾ by 7 to 11, eighty-three, 20 of which are 7 by 9 and 17 are 7 by 10.

8 to 8¾ by 9½ to 12, sixty-four, 15 of which are 8 by 10½ to 11, and 16 are 8½ by 10½ to 11½.

9 to 9¾ by 10¼ to 12½, thirty-three, 17 of which are 9 by 12.

10 to 11 by 11 to 14½, six.

The first four groups contain 80 per cent, and the last four groups only 20 per cent of the whole number of catalogues.

RECOMMENDED SIZES

| | Inches |
|---|---|
| **Index card, standard** | **3 x 5** |
| **Index card, larger sizes** | **4 x 6 and 5 x 8** |
| **Folders:** | |
| **Small** | **3⅝ x 6** |
| **Large** | **3¾ x 8½** |
| **Catalogues, standard size** | **6 x 9** |
| **Bulletins and large catalogues, standard** | **8½ x 11** |

In addition to the recommendation as to sizes the Committee would also recommend the following:

*a* For paper-covered catalogues intended to be permanently filed, the edges, including the cover, should be trimmed to exact size. No fancy deckled-edge or dark-tinted paper should be used.

*b* Overlapping edges of the cover are permitted when the catalogue is bound in covers stiff enough to support its weight in boards or heavy card paper.

*c* The title should always be printed on the exposed back of the catalogue whenever possible, for the purposes of identification. When the book or catalogue stands on its lower edge the title should be read from the top downwards, as in Fig. 1.

*d* Every catalogue should have a date on its title page.

*e* A standard size index card 3 in. by 5 in. should be enclosed in each catalogue, with the title of the book and a brief statement of the character of its contents printed on it.

| | |
|---|---|
| Catalogues, extra large, and technical press | 9x12 |
| Transactions of societies, pamphlets, etc | 6x9 |

PAPER BOXES, DRAWERS AND VERTICAL FILING CABINETS
**(Inside Measurements)**

| | |
|---|---|
| For 6-in. by 9-in. catalogues, pamphlets and 9-in. by 12-in. papers folded once | 6½x 9¾ |
| Paper boxes and drawers for larger sizes | 9½x12⅛ |
| Vertical files for large sizes | 10x12⅛ |

Fig. 2, which gives a clear idea of the relation of these several sizes to each other is drawn to scale, showing them superposed.

FILING BOXES AND CABINETS

*Jesse Jones Paper Box Company, Philadelphia, Pa.* A box that will hold a sheet $8\frac{1}{2}$ in. by 11 in. is by far the most in demand.

Various makers of filing cabinets give the following sizes, in inches, of drawers (inside measurements):


FIG. 1 SHOWING ARRANGEMENT OF TITLE ON BACK OF BOOK

**Correspondence files, $9\frac{7}{8}$ by $11\frac{9}{16}$, 10 by 12, $10\frac{1}{2}$ by $12\frac{1}{8}$, $10\frac{7}{16}$ by 12.**
**Legal cap files, $9\frac{7}{8}$ by $14\frac{3}{8}$, $9\frac{1}{2}$ by 15, $10\frac{1}{2}$ by $15\frac{1}{4}$.**
**Invoice files, $7\frac{1}{2}$ by $9\frac{1}{2}$, $7\frac{3}{4}$ by $9\frac{3}{4}$, 8 by 10, $8\frac{1}{4}$ by $10\frac{1}{16}$.**

The $9\frac{7}{8}$ in. by $11\,{}^{9}/_{16}$ in. is too short to take a 9-in. by 12-in. paper. There seems to be no reason for making the height more than 10 in. as no letter paper is over $8\frac{1}{2}$ in. wide. A size of 10 in. high by $12\frac{1}{8}$ in. wide would seem to be the best for all large-size catalogues. It would hold two 6 in. by 9 in. side by side, although the filing of 6-in. by 9-in. and 9-in. by 12-in. papers together in the same

drawer is objectionable, as the smaller sized papers have a tendency to lap over on each other and thus to occupy more space than they should. The extra inch in height gives ample room for division cards and tabs. The legal cap and the invoice files are not suitable for any of the standard sizes of catalogues.

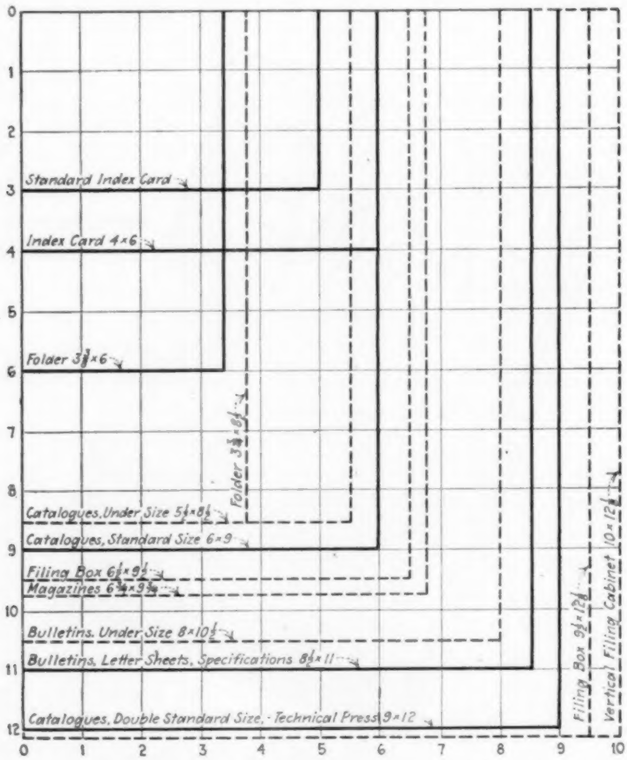



**Fig. 2 Relation of Catalogue Sizes**

A size suitable for the 6-in. by 9-in. catalogue does not appear to be in the market. If one were made with drawers about 6½ in. by 9¾ in. it should have a ready sale. The drawer would be more easily handled than the 10 in. by 12⅛ in. size.

Respectfully submitted,

| | |
|---|---|
| Wm. Kent, *Chairman*<br>J. R. Bibbins<br>M. L. Cooke<br>W. B. Snow | *Committee on Standardization of Catalogues* |

## No. 1395

# THE GERMAN MEETING

In June and July 1913, the official party of The American Society of Mechanical Engineers attended the Annual Meeting of the Verein deutscher Ingenieure at Leipzig, Germany, and participated in a remarkable tour of industrial Germany at the invitation of the Verein. The inception of this trip was at the time of a visit to this country of a study commission of the German Museum, headed by Dr. Oskar von Miller. Dr. von Miller as President of the Verein deutscher Ingenieure extended the invitation in person, and on June 8, 1912, the official invitation was confirmed by the governing body of the Verein deutscher Ingenieure. Subsequently, Prof. Conrad Matschoss, Dozent of the Royal Technical High School, Berlin, visited this country, officially accredited to arrange the details of the meeting, and after a conference with the members of the Council and the Committee of Arrangements, preliminary plans were made. Professor Matschoss also visited the several cities in Germany that were to be included in the trip and completed arrangements, and a wonderfully diversified program was laid out which included the itinerary published herewith, following which is the report of the meeting presented to the Council by the Chairman of the Committee of Arrangements. The Verein further provided for the comfort of every one by assigning Professor Matschoss to conduct the party through Germany.

## ITINERARY

### PLYMOUTH

Representatives of the Verein deutscher Ingenieure boarded the steamer to welcome the visitors.

### HAMBURG

*Thursday, June* 19

Arrival in Hamburg.

*Friday, June* 20

10.30 a. m. Breakfast in the Landungsbrücken-Restaurant.

11.00 a. m. Lecture on the harbor of Hamburg and the Elbe Tunnel, followed by an excursion about the harbor.

5.30 p. m. Visit to the Town Hall and reception by the Senate. Evening reception of welcome in the Ratsweinkeller by the Hamburg Section of the Verein.

*Saturday, June* 21

Excursions to the Elbe Tunnel, the shipyards of Blohm & Voss, and on the Hamburg Elevated Railway to the Power Station, Barmbeck.

In the afternoon, excursion by elevated railway and tram to Hagenbeck's zoological garden.

*Sunday, June* 22

Departure for Leipzig.

LEIPZIG

8.00 p. m. Evening reception of welcome at the Krystallpalast.

*Monday, June* 23

9.15 a. m. Opening session of the General Meeting in the Central Theater.

During the morning the ladies visited the Town Hall, the Imperial Court of Appeals, the University Library, and attended a short concert in the Merchants' Union Concert Hall. Luncheon in the Central Theater.

4.30 p. m. Concert in the Gewandhaus tendered by the Senate of the City.

6.30 p. m. Banquet in the large Festival Hall of the Central Theater.

*Tuesday, June* 24

9.15 a. m. Scientific lectures in the congress room of the Architectural Exhibition.

During the morning the ladies visited the Museum of Fine Arts and the Festival Hall of the University, followed by an automobile trip through the city.

1.00 p. m. Dinner in the exhibition restaurant, followed by an inspection of the exhibition.

4.30 p. m. Visit to the Monument of the Battle of the Nations. Address by Kammerrat Thieme, President of the Patriotic Union. Also technical excursions to the Machine Works of Karl Krause, the Leipzig Machine Tool Factory Company, Oscar Brandstetter, printers, and J. J. Weber, Illustrirte Zeitung.

8.00 p. m. Festival in the Palm Gardens.

*Wednesday, June* 25

9.20 a. m. Departure for Dresden.

DRESDEN

2.00 p. m. Rendezvous at the Belvedere, automobile excursion to Saxon Switzerland (Bastei), walk to Rathen, steamboat trip to Pirna and return to Dresden in automobiles.

8.30 p. m. Reception in the Rokokosaal of the Exhibition Palace, given by the City of Dresden.

*Thursday, June* 26

9.30 a. m. Excursions to the following points: Mechanical engineering laboratory of the Technical High School; sewing machine and bicycle factory of Seidel & Naumann; Picture Gallery and the Green Vault.

12.30 p. m. Luncheon at the Central Station, given by the Dresden Section of the Verein.

2.15 p. m. Departure for Berlin.

BERLIN

8.00 p. m. Reception in the Kaisersaal of the Rheingold.

*Friday, June* 27

9.30 a. m. Excursions to the following points: General Electric Company; Oberspree Cable Works; Siemens & Halske Company's Wernerwerk and the dynamo factory of the Siemens-Schuckert Works, Ltd.; Bergmann Electrical Works Company; Ludwig Loewe & Company; A. Borsig Works.

8.00 p. m. Banquet in the Marmorsaal of the Zoological Garden.

*Saturday, June* 28

2.00 p. m. Rendezvous at the Königsplatz. Automobile drive along the Heerstrasse to Wannsee and steamer trip on the Havel lakes.

7.00 p. m. Closing ceremony with supper in Wannsee.

*Sunday, June* 29

8.00 a. m. Departure for Düsseldorf.

DÜSSELDORF

8.00 p. m. Reception in the Städtische Tonhalle, given by the City of Düsseldorf. Lecture on the Importance of the Rheinsh-Westphalian Industry.

*Monday, June* 30

9.00 a. m. Excursions to the following points: German Machine Works Company; Friedrich-Alfred Steel Plant of the Fried. Krupp Company; machine factory of the Corporation of Thyssen & Company; Gutehoffnung steel plant; Rhenish steel works. Visit by the ladies to the public welfare works of Friedrich Krupp Company.

2.00 p. m. Visit to the Duisburg-Ruhrort Harbor.

8.00 p. m. Banquet in the Städtische Tonhalle of Düsseldorf, given by the Rhenish Westphalian Committee.

*Tuesday, July* 1

9.00 a. m. Excursions to the following points: Haniel & Lueg; Ernst Schiess Company; German Machine Works Company, Benrath works; steel works of the Becker Company; the Bergische Land. Automobile ride through the city for the ladies.

4.30 p. m. Departure for Cologne.

COLOGNE

8.00 p. m. Rendezvous in the Zoological Garden. Entertainment given by the Rhenish Westphalian Committee.

*Wednesday, July* 2

9.00 a. m. Excursions to the following points: Dye Works of Friedrich Bayer & Company; Deutz Gas Engine Works; Humboldt Mechanical Engineering Works. Visit by the ladies to the Cathedral and city in automobiles.

8.00 p. m. Reception in the Gürzenzich, given by the City of Cologne.

*Thursday, July 3*

8.45 a. m. Departure for Coblenz. From there, trip by steamboat on the Rhine to Rüdesheim, and by rail to Frankfort-on-the-Main.

FRANKFORT-ON-THE-MAIN

8.00 p. m. Rendezvous with the members of the Frankfort Section of the Verein deutscher Ingenieure in the Palm Gardens.

*Friday, July 4*

11.30 a. m. Reception of welcome by the Municipality in the Römer, followed by luncheon in the Halls of the Römer, given by the city and the Frankfort Section of the Verein.

3.00 p. m. Excursions: the East harbor and the gas works; slaughter house and the high reservoir; refuse destroyer plant and filters; visit to the old town and the Goethe House; Adler Works; Frankfort Mechanical Engineering Company; Simon, Bühler & Baumann; Voigt & Haeffner Company; visit to the Saalburg.

In the evening Independence Day celebration of the American colony in Homburg, v.d.h.

*Saturday, July 5*

10.15 a. m. Departure for Mannheim.

MANNHEIM

12.30 p. m. Luncheon in the Freidrichs Park, given by the firms visited.

2.30 p. m. Excursions: Machine Factory, Heinrich Lanz; Machine Factory, Sulzer Brothers; Brown, Boveri & Company; Portland Cement Works, Heidelberg & Mannheim Company; Benz & Company, Rhenish Automobile and Motor Company; the harbor and the Ludwigshafen Grist Mill Company; Carriage drive for the ladies through the city; visit to the Museum of Art, followed by tea.

8.00 p. m. Reception of welcome in the Nibelungen Saal of the Städt Rosengarten, given by the City of Mannheim.

*Sunday, July 6*

11.00 a. m. Excursion to Heidelberg.

HEIDELBERG

12.00 m. Luncheon in the City Gardens, given by the Mannheim and the Pfalz-Saarbrück Section of the Verein.

2.00 p. m. Visit to the castle.

4.00 p. m. Concert and tea in the Castle Restaurant.

6.30 p. m. Supper at the "Molkenkur," given by the Mannheim and the Pfalz-Saarbrück Section of the Verein.

9.15 p. m. Excursion on the Neckar to view the illumination of the castle with fireworks, given by the City of Heidelberg.

11.30 p. m. Return to Mannheim.

MANNHEIM

*Monday, July 7*

10.00 a. m. Departure for Munich.

MUNICH

8.00 p. m. Reception of welcome by the Bavarian Section of the Verein in the Hofbräuhaus.

*Tuesday, July 8*

9.30 a. m. Visit to the German Museum and the New Museum Building.

12.00 m. Luncheon in the New Museum Building, given by the German Museum.

3.00 p. m. Excursion to the Lake of Starnberg.

8.30 p. m. Closing ceremony. Banquet, given by the City of Munich in the Old Town Hall.

## REPORT OF GERMAN MEETING COMMITTEE[1]

The Committee began its activities with the general circular of August 27, 1912, followed by twelve other circulars and eleven postal cards giving full particulars of the program and itinerary. After careful search among available steamers, the great steam yacht Victoria Luise, 17,000 tons, of the Hamburg-American Line, was selected, and the travel bureau of this same line chosen to make all arrangements for the land trip of twenty days. As the steamer could carry about 600 passengers our right of choice of rooms was limited to March 1, 1913, but later extended a short period. By February 19, 209 had registered in the official party for the land tour in Germany, and we were informed that our final limit was fixed at 200 engineers and 100 ladies, as neither the special trains nor the hotels in Germany could take more than 300 in all. From the official lists there were 261 scheduled to sail on the steamer and 42 to join the party at Hamburg, making a total of 303. Owing to my long and repeated absences from New York most of the preliminary work was done by the other members of the Committee under the able and untiring leadership of Mr. J. W. Lieb, Jr., the Vice-Chairman.

We sailed from Hoboken at 10:10 a.m. on Wednesday, June 10, 1913. Secretary Rice and I were the only members of the original committee on board. Under authority given me by the Council I increased the committee to 17, making, with the Secretary and the Editor, 19 in all, as follows:

[1]Presented at the Council meeting, October 1913.

EXECUTIVE COMMITTEE

E. D. Meier, *Chairman*
W. R. Warner, *Vice-Chairman*
J. R. Freeman, *Vice-Chairman*
James Hartness
Jesse M. Smith

COMMITTEE ON EXCURSIONS

J. G. Brill, *Chairman*
L. P. Breckenridge
A. M. Greene, *Vice-Chairman*
H. M. Leland
E. E. Keller

COMMITTEE ON RESOLUTIONS

Sen. Newell Sanders, *Chairman*
H. M. Leland
F. R. Low, *Vice-Chairman*
F. G. Kretschmer
Wm. A. Doble

COMMITTEE ON SPEAKERS

H. L. Gantt, *Chairman*
Jesse M. Smith
H. G. Reist, *Vice-Chairman*
L. W. Nelson
E. E. Keller

The Committees on Entertainment, Acquaintanceship and Fourth of July Celebration had been appointed some time before and were already at work. The Entertainment Committee led by Prof. and Mrs. A. M. Greene, Jr., had provided 19 principal events.

But the steamer was steady and the sea smooth so that the entire program was enjoyable. The other passengers were invited to join in our exercises, by individual invitations sent to their cabins. They entered into the spirit of the occasion with alacrity and at the close of the trip thanked us in a poem posted on the bulletin board, signed by all individually. On June 15 the captain gave a special dinner in honor of the 25th anniversary of the accession to the throne of Emperor William II at which our congratulatory address was read. To this a German orator responded with much feeling. The resolutions were sent to Berlin by aerogram next morning.

On our arrival at Plymouth at 5 o'clock in the morning of June 18, Professor Matschoss, representing the Verein deutscher Ingenieure, and Engineer Kroebel, president, and Director Molsen and Professor Frasch, members, of the Hamburg branch of the V. D. I., came aboard to welcome us to Europe, and three officials of the Hamburg-American Travel Bureau opened an office on board to issue all the necessary

circulars and tickets to the entire party. After breakfast President Kroebel delivered his address of welcome in such fervent and ardent German that its spirit was fully understood by all in our party and a cordial friendship created which thenceforth grew with every new committee we met, and they were many. These were: (1) The council of the V. D. I., of members; (2) the honorary reception committee, 52 members; (3) eight local reception and entertainment committees, 329 members.

While naturally many of these were members of several committees, they included not less than 300 men prominent in government, manufacturing, railways and various branches of engineering, and as such they bore to our visiting party the welcome of the Verein of more than 24,000 engineers, of the vast and varied industries of Germany, of ten municipalities, of six sovereign states, and of the great German Empire.

It appears that when the V. D. I. had received our acceptance of its invitation a large number of its forty-eight branch societies applied for the privilege of receiving and entertaining us. With just regard to the time and strength of our party, the Council finally selected eight local branches; no sooner was this done than the ten cities covered by these branches demanded their right to participate in the welcome. Forty-six excursions to prominent engineering works and industrial concerns were arranged, explanatory lectures prepared, the transportation and commissariat arranged, and all with that German thoroughness and forethought that made it possible to enjoy everything with the greatest benefit and least fatigue.

The principal events were of course the two official joint meetings with the V. D. I. at Leipzig. The first, on June 23, after the official welcome, was devoted to two papers: one by Privy Councillor Lamprecht, on The Technic and Culture of the Present Day; the second, by Dr. W. F. M. Goss, President Am. Soc. M. E., on Foundations of American Engineering. Both treated their subjects in a broad and thorough manner, and it was instructive and gratifying to hear how the conclusions reached testified to the solidarity of engineering thought in the two great nations.

The King of Saxony with a brilliant court was an attentive listener for a full hour and was duly promoted to Doctor of Engineering. Our honored past-president, George Westinghouse, was unanimously voted the gold Grashof medal for his contributions to engineering science. A certificate accompanied this which contained the following

inscription: "Presented to Mr. George Westinghouse, who opened up new fields by his invention of the automatic railway brake, successfully fought for the introduction of the alternating current in the United States, and did useful work in the designing of high-speed machinery." President von Miller then presented an artistic plaque to one Society commemorative of the occasion, which now hangs in the rooms of the Society and reads as follows: "We salute the members of The American Society of Mechanical Engineers in the most cordial manner as the guests of our fifty-fourth general assembly in Leipzig. With a high respect for the scientific achievements and the good works of the American engineers we combine the warm and friendly feeling which arises from the common work devoted to the good of humanity in the field of engineering embracing all the civilized nations."

The second meeting, on June 24, was devoted to industrial efficiency. The subject was introduced by a comprehensive paper by James M. Dodge, Past-President, followed by an equally thoughtful one by Professor Schlesinger of the Technical University of Charlottenburg. An interesting and warm discussion followed in which the original, scientific and humanitarian work of our Past-President Taylor was duly acknowledged as fundamental and leading. The itinerary of our trip has been fully published. Having our special train always at our disposal and the professional promptness of our party harmonizing with the perfect arrangements of the travel bureau, we were able to carry out the program as published in advance, although it included in addition to the study of the plans freely opened for our inspection, our grateful acceptance of many social functions artistically arranged and gracefully tendered, numbering no less than fifteen formal banquets and receptions, fourteen luncheons and collations, twelve concerts, lectures and special performances, and ten excursions by rail and boat. In all of these there were present some 200 or more of our German hosts. In several we had occasion to admire the democratic attitude of German officialdom at social functions. Thus, the welcome to Berlin in a carefully prepared speech by a high official of the imperial ministry of the interior, the polite comradeship of Prince William of Saxe-Weimar at Heidelberg, the visit of the aged Prince Regent Ludwig of Bavaria to our excursion steamer on Lake Starnberg in the pouring rain, with an hour's friendly chat on canals, and the uniform urbanity of the lord mayors.

The social functions had their own distinctive features in each city. There was manifest an amicable rivalry in giving us the best

they could devise, especially as suggested by some local or historical fact, and worked out with the aid of good home talent in the arts employed. To mention any of these without a full description in this short report would savor of want of appreciation of the beauty and meaning of the performances. Full details are available in the letters, printed records, and photographs in the hands of the Secretary.

It is my pleasant duty to record the conscientious and telling labors of our various committees which so thoroughly coördinated the events that there was not the slightest hitch anywhere. The value of this was brought home to me by the Consul General of the United States in Berlin, when I called on him some weeks later. He said that during the past twelve months, two large American parties visiting Germany had inadvertently given serious offense by some omission, but that he had received from various cities and officials such gratifying reports of our party that these incidents were now forgotten and the American engineers accepted as the true representatives of our people.

Praise is due also to a group of graduates and undergraduates of several of our technical colleges for their untiring work as messengers on our trains and in hotels in distributing papers and packages, calling meetings and giving information.

We were all deeply impressed by these facts concerning our German hosts: That they believe laws are made to be obeyed, and though they are not blessed with such myriads of laws as we, those they have they respect; that the discipline learned in school and away has built up an industrial prowess capable of even greater conquests than those of the present; that the builders of the great empire desire peace to enable them to continue the beneficent battle compelling all natural force to the service of man; that nowhere has the engineer won a higher social standing than as accorded him in Germany; and that his ethical sense and ideals are the same in all countries and on fully as high a level as those of the older professions.

Respectfully submitted,

E. D. Meier, *Chairman*

No. 1396

# MEETINGS JULY-DECEMBER

## MEETINGS IN LOCAL CENTERS

### ATLANTA, JULY 12

Organization meeting, under the auspices of the Society, attended by representatives of the American Institute of Architects, the American Society of Civil Engineers, the American Institute of Electrical Engineers, the American Chemical Society, and the Engineering Association of the South, with a number of addresses. A full account is published in The Journal, September 1913.

### MILWAUKEE, OCTOBER 8

Address: The Uses of Pitot Tubes for Measuring Air, A. G. Christie. Based on the paper on Pitot Tubes for Gas Measurement, by W. C. Rowse, published in this issue of Transactions.

### NEW YORK, OCTOBER 14

Paper: Stability in Flying Machines, A. A. Merrill, lecturer on Aeronautics at Massachusetts Institute of Technology. Published in The Journal, October 1913.

### ST. LOUIS, OCTOBER 15

Paper: The New Turbine Pumps at the Chain-of-Rocks, L. A. Day, mechanical engineer, St. Louis Water Department. Discussion by E. H. Ohle, John Hunter, E. H. Tenney, G. M. Peek, and others.

### CINCINNATI, OCTOBER 16

Paper: Stresses in Machine Frames, A. L. Jenkins.

### WORCESTER, OCTOBER 17

Group meeting under the auspices of the Boston local committee, including technical excursions, a lecture and demonstration of Aeroplane Propeller Experiments at the Worcester Polytechnic Institute, by David L. Gallup, an inspection of the works of the Norton Company, with three short talks—Artificial Abrasives, by Aldus C. Hig-

gins, the Manufacture of Modern Grinding Wheels, by Carl F. Dietz, and the Use of Grinding Wheels, by Charles H. Norton—and a dinner at the New Bancroft Hotel, with speeches by James Logan, ex-Mayor of Worcester, and Charles D. Washburn of Washburn & Moen. An account of the meeting is given in The Journal for November 1913.

### ST. PAUL—MINNEAPOLIS, NOVEMBER 5

Organization meeting, with election of officers.

### NEW YORK, NOVEMBER 11

Paper: A New Centrifugal Pump with Helicoidal Impeller, C. V. Kerr. Published in The Journal, October 1913. Discussed by W. F. M. Goss, C. G. DeLaval, H. S. Hillman, Frederick Ray, Selby Haar, W. A. Shoudy, J. H. Lawrence.

### SAN FRANCISCO, NOVEMBER 17

Election of committee for 1913-1914.

### BOSTON, NOVEMBER 19

Under the auspices of the Boston Society of Civil Engineers. Paper: Engineering Lessons from the Ohio Floods, John W. Alvord of Chicago. A more complete account appears in The Journal, December 1913.

### CHICAGO, NOVEMBER 19

Dinner meeting. Informal talk on Iron and Steel Industry of Chicago, Wm. A. Field, general superintendent, Illinois Steel Company. Remarks by R. W. Hunt, E. M. Hagar, president, Universal Portland Cement Company, H. J. K. Freyn. A more complete account appears in The Journal, December 1913.

### PHILADELPHIA, NOVEMBER 19

Joint meeting with Franklin Institute. Paper: Producer Gas from Low-Grade Fuel, R. H. Fernald, describing investigations made by the U. S. Bureau of Mines. A more complete account is published in The Journal, December 1913.

### CINCINNATI, NOVEMBER 20

Joint meeting with Engineers Club of Cincinnati. Address:

Stellite, Elwood Haynes of Kokomo, Ind. An account is given in The Journal, December 1913.

### NEW HAVEN, NOVEMBER 21

Quarterly meeting with afternoon and evening sessions. Papers: Coöperative Industrial Research at the Sheffield Scientific School with Connecticut Manufacturers, L. P. Breckenridge; Research Work of the Bureau of Mines, O. P. Hood; Coöperation of State and University for Industrial Research, A. N. Talbot; Oxy-Acetylene Welding applied to Boiler Seams, Henry Cave, Autogenous Welding Equipment Company, Springfield, Mass.; Safety Devices used in connection with Grinding Wheels, R. G. Williams; Accident Prevention in Europe and the United States, A. D. Risteen; History of the Manufacture of Brass, W. B. Edwards; Experiments with Residence Heating Boilers at the Mason Laboratory, D. B. Prentice; Motor Car Testing, E. H. Lockwood. A more complete account appears in The Journal, December 1913.

### PHILADELPHIA, DECEMBER 10

Topic: How far shall judgment be exercised in the interpretation of engineering specifications? Papers by L. H. Kenney, Morris L. Cooke, J. B. Lichtenberger, lawyer and lecturer on Business Law, University of Pennsylvania. Discussed by Calvin W. Rice, Henry Hess, J. P. Jackson, H. H. Quimby, T. C. McBride, and others.

### ST. LOUIS, DECEMBER 13

Annual meeting. Addresses by R. H. Tait, John Hunter, Mr. Gustafson, superintendent of the Rankin School of Mechanical Trades, describing his investigations of important trade schools abroad, and H. Wade Hibbard. A more complete account appears in The Journal, January 1914.

### BOSTON, DECEMBER 18

Joint meeting under the auspices of the American Institute of Electrical Engineers. Illustrated Lecture: A Great Engineering Disaster and the Lessons learned therefrom, B. A. Behrend.

## THE ANNUAL MEETING

The thirty-fourth Annual Meeting, held December 2 to 5, was one of the largest in the history of the Society, with a total registration of 1271, of which 778 were members. Following the plan of the last

two years, the professional sessions were to a considerable extent in charge of sub-committees appointed by the Committee on Meetings, with the result that there were several symposiums on widely diversified subjects which gave a very broad scope to the meeting. The chief social event was the German Dinner on Thursday evening, planned to give those who were not of the party visiting Germany last summer an idea of the hospitality and unusual privileges shown the visitors, as well as to make an enjoyable reunion for those who spent the time together during the trip. Of very unusual interest, also, was the lecture by John W. Lieb, Jr., on the life and accomplishments of Leonardo da Vinci, one of the most remarkable lectures ever listened to by those in attendance at an annual meeting.

Another pleasant social feature of the meeting was the tea and musicale given in the Society rooms on Wednesday afternoon under the auspices of the Ladies' Reception Committee.

The program follows:

## PROGRAM

*Tuesday Evening, December 2*

Opening session. President's address: Efficiency in Technical Education a Factor in the Development of Professional Ideals, W. F. M. Goss. Report of tellers of election of officers and introduction to the President-elect.

Reception by the Society to the President, President-elect, ladies, members and guests.

*Wednesday Morning, December 3*

BUSINESS MEETING

Reports of the Council and Standing Committees. Amendments to the Constitution. Reports of Special Committees on Flanges, Standards for Hose Couplings, and Pipe Thread Gages. New business.

PROFESSIONAL SESSION

NOTES ON THE FURTHER OPERATION OF LARGE BOILERS OF THE DETROIT EDISON COMPANY, J. W. Parker.

Discussed by R. H. Danforth, Alex. Dow, A. A. Cary, I. E. Moultrop, G. B. Preston, R. D. DeWolf, Wm. Kent, R. P. Bolton, D. S. Jacobus, D. M. Myers, Harrington Emerson, A. A. Straub.

TASK SETTING FOR FIREMEN AND MAINTAINING HIGH EFFICIENCY IN BOILER PLANTS, Walter N. Polakov.

Discussed by William Kent, D. S. Jacobus, H. G. Stott, A. A. Cary, R. J. S. Pigott, E. A. Uehling, C. A. Austrom, Harrington Emerson, F. B. Gilbreth, R. D. DeWolf.

THE PROPERTIES OF STEAM, R. C. H. Heck.

REPORT OF SUB-COMMITTEE ON HOISTING AND CONVEYING.

Discussed by Spencer Miller, William Kent, D. M. Myers.

DYNAMIC BRAKING FOR COAL AND ORE-HANDLING MACHINERY, Clark T. Henderson.

*Wednesday Afternoon*

Excursions to various points of interest.

SIMULTANEOUS SESSIONS

RAILROAD

Papers contributed by the Sub-Committee on Railroads.

STEEL UNDERFRAME BOX CARS, George W. Rink.

STEEL UPPER FRAME BOX CARS, R. W. Burnett.

Discussed by H. H. Vaughan, B. D. Lockwood, H. W. Hibbard, W. F. Kiesel, Jr., E. G. Chenowith, C. A. Seley, O. C. Cromwell, W. S. Atwood, F. M. Whyte.

CEMENT

Informal discussion on various topics.

TEXTILES

Papers contributed by the Sub-Committee on Textiles.

COTTON CONVEYING SYSTEMS; THEIR SAFEGUARDS AGAINST FIRE, H. A. Burnham.

Discussed by A. W. Thompson, C. J. H. Woodbury, J. A. Stevens, E. V. French, C. H. Bigelow.

SPECIFICATIONS FOR FACTORY TIMBERS, F. J. Hoxie.

Discussed by E. M. Bates, A. M. Ernst, C. H. Bigelow, C. J. H. Woodbury, E. V. French, C. T. Plunkett, Hermann von Schrenk.

TEXTILE COST ACCOUNTING, C. B. Annett and C. F. Cunningham.

RECEPTION

Musicale and tea given to the members and their guests by the Ladies' Committee in the rooms of the Society.

*Wednesday Evening*

Presentation of the Grashof Medal by the Verein deutscher Ingenieure to George Westinghouse, Past-President and Honorary Member of the Society.

Address, illustrated by lantern views, on LEONARDO DA VINCI, ENGINEER AND ARTIST, by John W. Lieb, Jr., Past Vice-President of the Society.

*Thursday Morning, December 4*

Excursions to various points of interest.

SIMULTANEOUS SESSIONS

GENERAL

EFFICIENCY OF ROPE DRIVING AS A MEANS OF POWER TRANSMISSION, E. H. Ahara.

Discussed by H. G. Stott, G. N. VanDerhoef, W. H. Kenerson, Selby Haar.

COMPARATIVE TESTS OF THREE TYPES OF LINESHAFT BEARINGS, Carl C. Thomas, E. R. Maurer and L. E. A. Kelso.

Discussed by E. H. Ahara, William Kent, Selby Haar.

PITOT TUBES FOR GAS MEASUREMENT, W. C. Rowse.

Discussed by D. S. Jacobus, W. H. Carrier, C. C. Thomas, Leo Loeb, F. R. Still, G. F. Gebhardt, C. P. Crissey, A. G. Christie.

TESTS OF VACUUM CLEANING SYSTEMS, J. R. McColl.

Discussed by H. M. Grossman, C. R. Thurman.

THE ART OF ENAMELING, OR THE COATING OF STEEL AND IRON WITH GLASS, Raymond F. Nailler.

MACHINE SHOP PRACTICE

Papers contributed by the Sub-Committee on Machine Shop Practice.

GEARS FOR MACHINE-TOOL DRIVES, John Parker.

Discussed by John Riddell, F.V. McMullin, A. L. DeLeeuw, H. F. L. Orcutt, A. W. Thompson, E. H. Neff, F. DeR. Furman, C. R. Gabriel.

CAST-IRON FOR MACHINE-TOOL PARTS, Henry M. Wood.

Discussed by W. W. McKaig, A. L. Jenkins, T. D. West, F. R. Jones, E. H. Mumford, A. L. DeLeeuw, D. J. Riggs.

A RECORD OF PRESSED FITS, C. F. MacGill.

Discussed by J. E. Sweet, S. A. Moss, H. M. Lane, John Riddell, A. B. Carhart.

STANDARDIZING MACHINERY, Fred H. Colvin.

Discussed by L. D. Burlingame, F. DeR. Furman, A. B. Carhart, F. J. Miller, James Green, G. S. Walker.

GAS POWER SECTION

Papers contributed by the Gas Power Section.

A NEW PROCESS OF CLEANING PRODUCER GAS, H. F. Smith.

Discussed by F. R. Hutton, W. T. Magruder, R. H. Fernald.

PRESENT STATUS OF THE LARGE GAS ENGINE IN EUROPE, Prof. P. Langer.

Discussed by H. J. K. Freyn, F. Z. Nedden, R. H. Fernald, F. S. Giller.

*Thursday Afternoon*

Excursions to various points of interest.

Informal tea served by the Ladies' Committee in the rooms of the Society.

*Thursday Evening*

Annual Reunion and Reception, Hotel Astor.

German Dinner in the grand ball room, followed by illustrated lecture descriptive of the chief events of the German trip during the summer of 1913, by Worcester R. Warner. Dancing.

*Friday Morning, December 5*

FIRE PROTECTION SESSION

Papers contributed by the Sub-Committee on Fire Protection.

THE FIRE HAZARD IN TURBO-GENERATORS, G. S. Lawler.

Discussed by B. G. Lamme, I. E. Moultrop, P. M. Lincoln, F. E. Cardullo, H. G. Reist, Selby Haar, Albert Blauvelt, R. D. DeWolf.

EXTINGUISHING OF FIRES IN OILS AND VOLATILE LIQUIDS, Edw. A. Barrier.

Discussed by A. E. Cluett, J. S. Thomson, H. W. Appleton, L. H. Kunhardt, Albert Blauvelt, Gorham Dana, F. E. Cardullo.

A SYSTEM FOR THE CONTROL OF AUTOMATIC SPRINKLER VALVES, Fred J. Miller.

Discussed by G. I. Rockwood, J. P. Tolman, Gorham Dana, W. H. Kenerson, C. H. Bigelow, P. W. Power, L. H. Kunhardt, F. J. Bryant.

THE NEED OF MORE CARE IN THE DESIGN AND CONSTRUCTION OF ELEVATED TANKS, W. O. Teague.

Discussed by J. W. Ketler, B. A. Freeman, Bryan Blackburn, A. H. Hayes, Albert Blauvelt, G. A. Smith, C. S. Pillsbury.

FIRE PUMPS, Ezra E. Clark.

Discussed by A. B. Carhart, Albert Blauvelt.

*Friday Afternoon*

Excursions to various points of interest.

No. 1397

# EFFICIENCY IN TECHNICAL EDUCATION A FACTOR IN THE DEVELOPMENT OF PROFESSIONAL IDEALS

By W. F. M. Goss, Chicago, Ill.

President of the Society

Fifty-one years ago, the Congress of the United States passed an act providing for the establishment of colleges in each of the several states of the Union.[1] The passage of this so-called Land Grant Act, more than any other single event, constitutes the foundation of technical education in the United States. It aided in the establishment of the Massachusetts Institute of Technology, of Cornell University, and in due time of colleges or universities offering engineering courses in every state. Many states were slow in proceeding under this act and, some, which today have flourishing universities, had not then been admitted into the Union. As a consequence, technical education, as a national movement, did not have its beginning until the early seventies, or approximately forty years ago.[2]

The coming of the technical school, whether as a department of the state university or otherwise, was not in any large sense the result of active interest on the part of engineers. It came as the unfolding of an educational movement inspired and sustained by scientists and

[1]This "Land Grant Act," sometimes referred to as the "Morrill Bill," in honor of Representative, and afterwards, Senator Justin S. Morrill, of Vermont, who labored through several sessions of Congress to secure its passage, was signed by Abraham Lincoln, July 2, 1862. It granted to each state in the Union 30,000 acres of land for each senator and representative to which it was entitled in the Federal Congress, for the purpose of promoting "the liberal and practical education of the industrial classes in the several pursuits and professions in life." It has recently been referred to by a distinguished educator as "the greatest endowment of higher education ever made at one time by the act of any legislature."

[2]This statement does not disregard the fact that engineering education under private auspices began earlier than indicated. The Rensselaer Polytechnic Institute was founded in 1824, incorporated in 1826, and reorganized under its present name in 1849.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

educators.[1] It came as an expression of a nation's desire to emphasize the importance of science in its relation to the every day work of mankind. It came when our country's engineering industries were young, when engineering practice was guided by precedent rather than by mathematical analyses, and when those who were skilled in leadership were also skilled in the manipulative processes of their art. It came when the industries offered few inducements to a young man whose chief claim to their attention was the fact that he had studied books, and when the limitations of the employer often made it difficult for him to appreciate the qualifications of such an applicant. The practical men, therefore, expected the new school to train its students in practice rather than in theory, to do things rather than to know facts; they looked to the school for a preparation which should be kinetic rather than potential.

The technical school was not unsympathetic to this expectation. It gave early evidence of a desire to meet the reasonable demands of those to whom its graduates must go. It gave generous attention to courses in shop practice, in field work, in drawing and in designing. It introduced certain text-book courses, largely descriptive, and in many other ways it emphasized the more practical aspects of the engineer's work. By so doing, the school met an immediate need, gained for technical education the good will of the leaders of industry, and laid the foundations for a degree of popular support, which, in process of time, transformed the simpler beginnings of nearly half a century ago into one of the world's greatest undertakings in higher education.

But, in giving their response to the immediate needs of the industries, the early leaders in technical education rarely lost sight of the possibilities of science as a means to the development of correct theory. Under the guidance of such men as Rogers, Runkle and Thurston, the art of the teacher was so combined with processes common to practice, that even training in practical things became an important educational influence. Shop courses and drawing courses became new expressions of an educational function. The study of

[1]In connection with this statement it will be of interest to note that President Edmund J. James of the University of Illinois, in a paper entitled "The Origin of the Land Grant Act of 1862," has shown that Jonathan B. Turner, at one time professor in Illinois College at Jacksonville, Ill., deserves the credit of having been the first to formulate clearly and definitely the plans of a national grant of land to each state in the Union, for the promotion of agriculture and the mechanical arts, and of having organized and continued to a successful issue, the agitation that made possible the passage of the bill.

things practical was put upon so high a plane that the attention of the student was secured, his power of observation increased, his view broadened and all his conceptions stimulated. Moreover, while admitting courses of a practical nature, the early leaders in technical education did not limit their scheme of instruction to such courses. They construed the federal law literally. They utilized the popular interest in more simple applications of science as an argument for the encouragement of work in more complex fields, and, as the usefulness of the men trained under these leaders became more and more apparent, an extension of their plan of work became possible. Courses of instruction were extended. Descriptive work gradually gave way to work which was more mathematical. Great laboratories for instruction and research were established, until the American student finds at home educational facilities for which he formerly went abroad.

Meanwhile the every-day citizen, who is the normal supporter of the great state university, is coming to understand something of the breadth of the educational problem which is presented by the activities of the technical school. He has seen the number of its students multiply, and its graduates become leaders in the work of great industries. Questions affecting the support of the technical school have sometimes appeared as political questions, and in his discussions of these the normal citizen has not been found lacking in his appreciation of the technical school. Engineering education now suggests something more to him than a class-room. He understands that efficient instruction must find expression in the applications of science and in the promotion of scientific research. These are important conceptions, and the fact that they are entertained by the public suggests the great influence already exerted by the technical school.

The American technical school stands today on the threshold of great achievements. Its pioneer days are nearly over. The fruitage of two generations of educational effort is now in evidence. Captains of industry no longer look in doubt upon the technical graduate; they give him work. Public confidence in the new education has been won and its permanent support is assured. What of the future? To what new attributes must the technical school give expression and what new responsibilities should it assume? These are questions which today confront the school, and they are questions also in which the membership of The American Society of Mechanical Engineers may

well be interested, for the technical school and the professional society are partners in a common cause.

An attempt to outline the problems of the technical school must recognize the fact that great advances in any art or profession await the coming of great men. The modern need in the field of engineering is for men who can perform the exceptional task; for men who are safe keepers of their brothers' interests; for men whose qualities of character are so sound and strong that they instinctively perceive the way of truth and follow it; for men whose activity and understanding detect the defects in established practice and find a way to improve that practice; for men who so well understand the fundamental principles of science as to be able to predict the effect of the next step before it has been taken; for men, in fine, who can rank with the world's great leaders in science and industry. I conceive it to be the prime purpose of the technical school to make its contribution toward the development of such men.

It would seem unnecessary to remind an audience of professional engineers that the achievements of the technical school center in the quality and strength of its instructional staff. The present-day need is for increased strength of staff. All honor is due the fine coterie of men who, undertaking the work of the technical school a quarter of a century or more ago, have adhered to their task in the face of discouragement at great personal sacrifice and with a spirit that has been fine and even heroic. But retrospect, however satisfactory in itself, will not solve our future problems. Great leaders in the work of the technical school are few, and the unfortunate fact is that throughout the years that are past few men have been in training where the service now demands many. There are many reasons for this. An important one is to be found in the limitations which have been put upon the salary budget. The man who builds great bridges or who directs the activities of a great industry, is none too gifted to guide the efforts of ambitious youth, and hold before the student large ideals of life. And yet, in the industries a man may receive in a month an amount equivalent to the annual salary of many college professors. It is clear to all who know the field that while the school must suffer some years to come, through its deficiencies in the past, a new policy should be entered upon as speedily as possible. It should be expressed in such terms as will convince the exceptional man, looking for a career, that it will be worth his while to prepare himself by study, by professional practice, by travel, by activities as a scientific investigator, by every process which can develop and broaden,

for a life-work as a member of a college faculty. Few among American technical schools can today offer professorships which in themselves are sufficiently attractive to justify a young man in securing for himself so elaborate a preparation. The remedy cannot ordinarily be applied by the college itself; it must come as the result of interest shown at the sources of financial supply. In the great state universities it must be the result of public appreciation of the need; you, I know, will realize that public appreciation, though a plant of slow growth, may be cultivated by many different agencies.

Not only is it required in the interest of higher efficiency that the American technical school have an instructional staff of the highest possible quality, but it is also required that such a staff be not overburdened with routine duties. The American youth is greatly influenced by the personality of his instructor. While the exceptional student will view his problems broadly, will add his own personality to that which the professor gives and thus work out large and vigorous conceptions, the normal student is as a disciple following a master; he admires the master's skill, he thinks in terms of the master's thoughts and is very likely to be influenced by the master's limitations. If the engineering graduate sometimes degenerates into an animated slide rule, may it not be possible that he has been instinctively led to such a career through ill-conceived tread-mill processes in the classroom? If so, the remedy is to be found in reforming the work of the classroom, and one sure road to such a reform is that which opens the way whereby men of large caliber may have time in which to impress themselves upon their students. The technical school is greatly in need of simple living and high thinking. Simple living is easily attained unto, but high thinking, in its essence, involves a certain element of leisure, or perhaps freedom in the choice of one's occupation, which after all, is but another term for leisure, and no atmosphere surcharged with high thinking can prevail on a campus where every individual student and professor is perpetually keyed up to concert pitch in an endeavor to accomplish an assigned task. This is a matter in which our English and German neighbors can, by the trend of their procedure, disclose much which will be serviceable to the American technical school.

The importance of superior leadership in the school is emphasized by the fact that the attitude of the American student reflects rather faithfully the changing spirit of the times. His parents in their home, though perhaps living unpretentiously, no longer pursue the quiet and simple ways of former days. They have their part in the

complexity of the modern business and social life. It is not surprising that their son, the student of today, finds some difficulty in settling down to the quiet routine of the cloister. He is more likely to combine with study the social and athletic activities of the college. The encouragement of fellowship with men, as well as of study, is in fact recognized by the school authorities as a legitimate function of the university. The danger appears in a disposition on the part of students to segregate their activities, to regard their study as work and their athletics as play. Comparisons growing out of such distinctions are to the disadvantage of the student's occupation. The habit of placing the emphasis continually on the joy of the play, prevents him from feeling the joy of the more intellectual pursuits and it encourages half-hearted study. It is here that masterful leadership in the classroom will assert itself. The joy of scholastic achievement is best made plain by the introduction and maintenance of inspiring views depicting the meaning of a life of service coupled with the conception that student days are part of such a life; that student days are not merely a preparation for life, but that they are life.

Masterful leadership eliminates entirely the feeling once entertained by the students, that the instructor is a taskmaker, and that to be obedient to him, the student must perform the task. The great teacher, freed from the burden of excessive routine, may easily recognize differences in ability and will encourage the student who must plod and inspire to unusual performance the brilliant student who knows no limit to his achievement save his physical strength.

Assuming the technical school to be in possession of an ideal instructional staff, the way will open for progress through many channels of secondary importance. It is altogether possible that in our present-day routine too much time is given to things which are simple. Much that is now studied may perhaps be read. The habit of studying intensely a few books to the entire exclusion of the great mass of historical and biographical engineering literature affords the student but little opportunity of acquiring a habit of rapid and intelligent reading, which in itself is an accomplishment worth striving for. The practice of the shop laboratory, the drawing-room, the surveying field and the study of descriptive texts, rightly interpreted, are important adjuncts in the training of the engineer, but the time has ceased to be when such activities constitute the chief characteristics of the technical school. Year by year the technical school has increased the emphasis given to processes which are mathematical. The progress of the next decade will be seen in the

thoroughness with which high standards in such work are accepted and advanced. The intensive work of the course must be based upon fundamental theory, and the fields to which such theory is applied must be broadened. The engineering graduate is no longer required to be prepared to operate machines, but he must have a well-trained mind and he must possess power to perfect his qualifications along any specialized lines in the shortest possible time. To this end, the years in college must be spent in acquiring an understanding of principles and in the development of those aspects of theory which are difficult to acquire after one's college days are over.

The aspirations of many students will not be satisfied by the possibilities of a four years' course, and for these graduate work must follow. In making this assertion I do not forget that in this country graduate work in engineering has thus far received but little attention, but this is due rather to our immaturity than to the fact that we do not recognize its need. The time is at hand when the services of the scholarly engineer, the man who through his perfect command of fundamental theory can visualize a cause from its effects, will be in great demand. The manufacturing industries, the great commercial laboratories of our country and the colleges themselves are in need of men who have done more work than any one can accomplish in the normal four years of a college course. The problem of graduate work in engineering is therefore a national one, and it is worthy of note that a number of universities and technical schools are now holding before the public visions of its value.

The American technical school has already accomplished much in the development of laboratories. While it has used them thus far chiefly as facilities for instruction, it has always recognized the fact that there are other and equally important functions to be performed by them. The laboratories of a considerable number of schools have made important contributions to the sum of human knowledge. This fact is suggestive of future possibilities. The practice of men must be guided by facts. The technical school cannot content itself with routine service. It must enshroud its classroom and its laboratories with an atmosphere of scientific achievement; its professors should be leaders not only as classroom instructors, but nation-wide leaders, even world-wide leaders in the complex and highly diversified fields of the science which they represent. The laboratories available for the use of such men should multiply. Every technical school should aid, as individual schools have already aided, in setting forth new interpretations of physical phenomena. The laboratory must be more

than a shop, more than an engine room, more than a collection of testing machines. No great laboratory can be ordered ready-made. It cannot be produced in response to a decree. Whatever its dimensions or its cost, it can be great only in so far as it reflects a purpose which is scientifically sound, and employs means which are scientifically correct. Its significance is necessarily limited to the qualities of the men who create and operate it. A laboratory which has been evolved through the activities and desires of a master is not only priceless for the school that possesses it, but necessary as a source of information of the highest value, to the field of practice which it is designed to serve. The chief engineer of a great railway system, in writing of the work of a certain college laboratory, has recently certified in terms which are clear and emphatic, to the value of principles affecting the design of certain structures which had been developed by the laboratory and extensively used by the corporation which he served. Illustrations of similar import will occur to all readers of technical literature. Assuming that the country has need of research laboratories for purposes quite apart from the work of instruction, it can, I think, be shown that the technical school constitutes the most promising agency in our national economy upon which to place the responsibilities incident to their creation and maintenance.

In concluding this phase of my discussion, I must not fail to recognize the contributions of members of this Society to the upbuilding of the American technical school. I do not forget that some of our members are university, college or technical school professors; that others are presidents, and still others are officers or members of administrative boards. They will readily understand, however imperfect my language, the significance of the matters which I am endeavoring to emphasize. To those of you who have a lay interest only in technical education, let me suggest that you do not always time your visits to your Alma Mater to make connection with a great football game or a great baseball game, but that you make some visits at a time when you can show interest in the academical progress of the student body; not because the game is unworthy, but that you may secure some understanding of the students' achievements in severely scholarly efforts, which after all is the measure by which the success of the school must ultimately be judged.

The relation existing between the standard of instructions in schools which prepare for a profession and the ideals of the profession has already been suggested and in its general aspects is obvious. The fact that educational institutions of high standing are

sending out each year into the engineering pursuits of our country more than 2500 graduates, nearly 1000 of whom enter the field of mechanical engineering, suggests the basis for this relationship. The technical school is, in fact, recruiting and otherwise stimulating the engineering work of the country. Its more important contributions may be summarized as follows:

First, the work of the school tends to emphasize the dignity of the calling. Professionalism as distinguished from the art or practice of engineering is after all a question of quality. Professionalism is breadth! In our ambition to have a large part in the world's affairs we sometimes forget the fundamental source of power. We sometimes think too much of honors and too little of service; we desire the position before we are qualified to discharge the responsibilities it imposes. Our strivings must be for fitness. Progress in advancing the ideals of our profession will depend upon the character of the service rendered by the members of the profession. If every practitioner in the field of Mechanical Engineering possessed character and ability responding to our highest ideals, no one would question the right of mechanical engineers to regard themselves members of a profession. The technical school makes its contribution to the upbuilding of professional ideals by sending trained men into the profession.

Second, a service which the technical school is rendering the profession of engineering is that of contributing to the sum of its scientific data. Engineering as a science has made progress by leaps and bounds; rule of the thumb has given way to the rule of the mathematician. But notwithstanding all that has been accomplished in this direction, there are as yet but few departments in the field of mechanical engineering in which the basis of design is actually perfected. Further progress awaits the establishment of facts concerning such matters as the behavior of constructive materials, of lubricating films, of combustible mixtures, and of liquids and gases employed in thermo-dynamic processes as vehicles for the transmission of heat. The American engineer who has hitherto been occupied with the so-called practical aspects of his profession has concerned himself less than have the engineers of the great empire across the sea, with the work of the scientist. We, as engineers, need to train ourselves to a condition of mind which will make studious processes less difficult than at present. Along this line of endeavor the researches of the technical school will aid the profession by establishing new

standards of proficiency, and by making new contributions to the existing fabric of facts.

Third, an important contribution which the school is rendering the profession of engineering is that of emphasizing the profession's unity of purpose. The college graduate has had at least four years' training in teamwork. A majority of the student organizations with which he has been associated has maintained extra-mural affiliations. The college fraternity, the college Y.M.C.A., the college athletic activities would lose much of their significance if their whole thought and action were restricted to the home campus. In intercollegiate contests the desire is not in any large sense to win over a neighbor, but to win as an achievement. A graduate of Wisconsin finds a friend when he meets for the first time a graduate of Illinois, and a member of the faculty of one institution, upon a trip of visitation to a neighboring institution, is sure to find himself among those who cheer and stimulate him. This intercollegiate spirit of fellowship has become a national asset—it persists when the day of graduation has passed. It represents a spirit which, in the business and professional world, conserves and upbuilds. It is working out the world's great problems of mutual respect, of mutual help and of concentration of purpose. In the development of our future ideals and practices, tradition and prejudice are likely to play a steadily diminishing part, and the spirit of fellowship, inherited from college days, a part of steadily increasing importance.

Whatever may be the nature of the contribution of the technical school to the ideals of the profession, the extent of the contribution is a function of the quality of the school. Better instruction in the school must supply better recruits for the profession, greater activities in scientific researches on the part of the school must operate to increase the facility with which the problems of the engineer are solved, and influences in the school which tend to extend the student's horizon and broaden his sympathies, will in due time make their impress upon the professional life outside of the school. That is, efficiency in technical education is a factor in the development of professional ideals, and hence a matter of prime importance to this society, as well as to all other organizations of engineers. I have presented this discussion because I believe that the problems of the technical school should not be left to the school master, for, broadly interpreted, they are not the problems of the school but the problems of the profession. As such they should, I believe, receive painstaking and persistent attention from this and other engineering societies.

No. 1398

# STANDARD THREADS FOR HOSE COUPLINGS

## REPORT OF SUB-COMMITTEE ON FIRE PROTECTION

In considering the problem of the adoption of a hose coupling thread, it became a question whether to advise specifications which would show the extreme of mechanical strength without reference to the preponderance of designs of a less theoretical value in general use, or to seek for the introduction of a threaded coupling, the characteristics of which would most closely accord with the majority class, and at the same time prove to be an intermediary of such capacity as to accommodate itself to interchange with a large proportion of couplings not exactly conforming to its dimensions.

Accepting the latter method of procedure as promising the widest measure of success, a committee of the National Fire Protection Association undertook a special investigation of existing conditions, using the report of a special committee under C. A. Landy, chairman, in 1891, as a basis. After securing additional data they became convinced of the practical value of the specifications named in that report, and submitted as a standard coupling for $2\frac{1}{2}$-in. hose, one showing a diameter of $3\,{}^{1}/_{16}$ in. over male end thread with $7\frac{1}{2}$ threads to the inch, by the use of which it was practically demonstrated that couplings ranging in outside diameter from $3\,{}^{1}/_{32}$ in. to $3\,{}^{5}/_{64}$ in., with either 7, $7\frac{1}{2}$ or 8 threads to the inch, could be so modified as to couple-up in service with this suggested standard, and thus render over 70 per cent of the $2\frac{1}{2}$-in. couplings known to be in use, conformable to the proposed standard at small expense as to time, money or labor.

In elucidation of the essential features of this standard it was deemed wise to formulate specifications covering $2\frac{1}{2}$, 3, $3\frac{1}{2}$ and $4\frac{1}{2}$-in. hose couplings, the inside diameters of which were to be in conformity with the sizes named, specific details relating to each of the standard sizes being shown in the printed specifications as follows:

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

SPECIFICATIONS FOR HOSE COUPLINGS

| | | | | |
|---|---|---|---|---|
| Inside diameter of hose, inches | 2½ | 3 | 3½ | 4½ |
| Number of threads per inch | 7½ | 6 | 6 | 4 |
| *Male Couplings* | | | | |
| Outside diameter of thread *finished*, inches | 3 1/16 | 3⅝ | 4¼ | 5¾ |
| Diameter at root of thread, inches | 2.8715 | 3.3763 | 4.0013 | 5.3970 |
| Clearance between male and female threads, inch | 0.03 | 0.03 | 0.03 | 0.05 |
| Total length of threaded male end, inches | 1 | 1⅛ | 1⅛ | 1⅜ |

The above are to be of the 60-deg. V-thread pattern with 0.01 in. cut off the top of thread and 0.01 in. left in the bottom of the valley in 2½-in., 3-in. and 3½-in. couplings, and 0.02 in. in like manner for the 4½-in. couplings and with ¼-in. blank end on male part of coupling in each case. Female ends are to be cut ⅛ in. shorter for endwise clearance, and they should also be bored out 0.03 in. larger in the 2½-in., 3-in. and 3½-in. sizes, and 0.05 in. larger in the 4½-in. size in order to make up easily and without jamming or sticking.

CONVERTING NON-STANDARD COUPLINGS

The fact that the National Standard has received the unqualified approval of all the leading organizations concerned with water supplies and fire departments, forms a strong argument for its early adoption in all localities. In order to demonstrate that the question of expense in changing over to the standard is less serious than is often imagined, the following suggestions, contemplating a gradual change from non-standard to full standard equipment, are submitted in the belief that the comparatively light cost of such a procedure should not delay so important and beneficial an improvement in any town or city. These suggestions are intended to apply to the period of transition which must of necessity precede complete standardization.

Considering first the 2½-in. hose couplings and hydrant outlets in general use, we suggest that:

*a* Contracts for new hydrants should specify that the nipples be equipped with the National Standard Hose thread

*b* Existing hydrant nipples should be replaced by standard nipples. This may readily be accomplished at comparatively small expense through the use of a special device or tool now on the market; or as a less satisfactory method, the nipples may be equipped with adapters having standard

thread on the outboard end. These adapters should be fastened in position so as not to be readily removable.

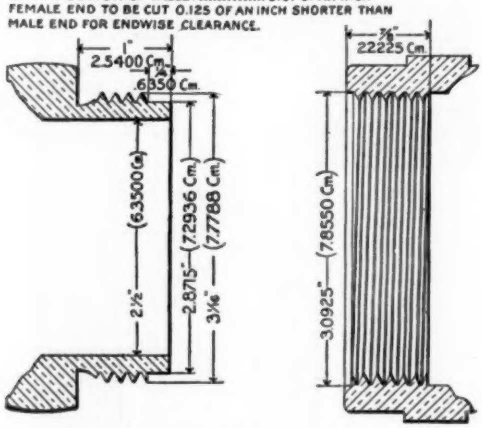
*c* Fire engine nipples should be provided with adapters having standard thread on the outboard end. These should be secured in place so as not to be readily removable.

*d* In many cities and towns where the 2½-in. hose couplings, as well as the nipples on hydrants and fire engine outlets,

NATIONAL STANDARD HOSE COUPLING

| | INCHES | CENTIMETERS |
|---|---|---|
| INSIDE DIAMETER OF HOSE COUPLING | 2.5000 | 6.3500 |
| BLANK END OF MALE PART | 0.2500 | 0.6350 |
| OUTSIDE DIAMETER OF THREAD FINISHED | 3.0625 | 7.7788 |
| DIAMETER OF ROOT OF THREAD | 2.8715 | 7.2936 |
| CLEARANCE BETWEEN MALE AND FEMALE THREADS | 0.0300 | 0.0762 |
| TOTAL LENGTH OF THREADED MALE END | 1.0000 | 2.5400 |

NUMBER OF THREADS PER INCH ......... 7½
PATTERN OF THREAD ......... 60° V
CUT OFF AT TOP OF THREAD ......... 0.01 OF AN INCH
LEFT IN BOTTOM OF VALLEY ......... 0.01 OF AN INCH
FEMALE END TO BE CUT 0.125 OF AN INCH SHORTER THAN MALE END FOR ENDWISE CLEARANCE.




2½" SIZE (6.3500 Cm.)

FIG. 1 DETAILS OF 2½-IN. COUPLING

show 7, 7½ or 8 threads to the inch, wide variations occur in outside diameter over the thread of the male end of the couplings. If such variation does not exceed $^1/_{32}$ in. below 3 $^1/_{16}$ in. (equaling 3 $^1/_{32}$ in.), or if the variation does not exceed $^1/_{64}$ in. in excess of 3 $^1/_{16}$ in. (equaling 3 $^5/_{64}$ in.), it becomes feasible to render both male and female couplings adaptable for interchange with the Standard 2½-in. hose couplings (measuring 3 $^1/_{16}$ in. outside diameter on the male end and 7½ threads to the inch) by the use of an adjustable tap for the female end

of the coupling, or an adjustable die for the male end of the coupling, either tap or die having the same number of threads to the inch as the coupling or nipple to be treated.

Any deviation within the limits named may readily and cheaply be overcome without the removal of couplings from the hose or of the nipples from the hydrant or engine. It

NATIONAL STANDARD HOSE COUPLING

| | INCHES | CENTIMETERS |
|---|---|---|
| INSIDE DIAMETER OF HOSE COUPLING | 3.0000 | 7.6200 |
| BLANK END OF MALE PART | 0.2500 | 0.6350 |
| OUTSIDE DIAMETER OF THREAD FINISHED | 3.6250 | 9.2075 |
| DIAMETER OF ROOT OF THREAD | 3.3763 | 8.5758 |
| CLEARANCE BETWEEN MALE AND FEMALE THREADS | 0.0300 | 0.0762 |
| TOTAL LENGTH OF THREADED MALE END | 1.1250 | 2.8575 |

NUMBER OF THREADS PER INCH.........6
PATTERN OF THREAD.........60° V
CUT OFF AT TOP OF THREAD.........0.01 OF AN INCH
LEFT IN BOTTOM OF VALLEY.........0.01 OF AN INCH
FEMALE END TO BE CUT 0.125 OF AN INCH SHORTER THAN MALE END FOR ENDWISE CLEARANCE.

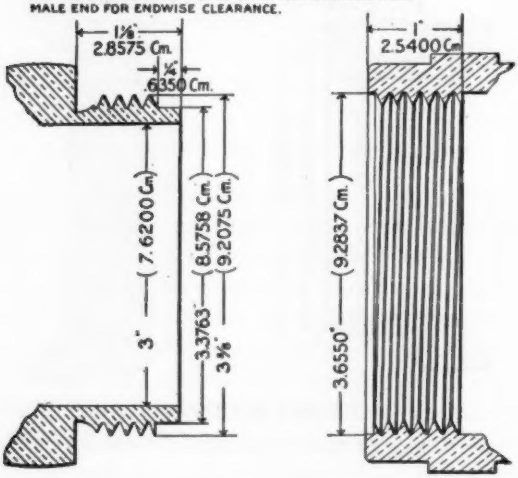



FIG. 2 DETAILS OF 3-IN. COUPLING

may be well to emphasize the fact that in adapting the 7 and 8-thread coupling to interchange with the National Standard of 7½ threads, it is thus intended to provide an interim measure to serve until the standard has been fully installed, the reduced coupling being discarded as the hose wears out, and all new hose purchased to be fitted with the

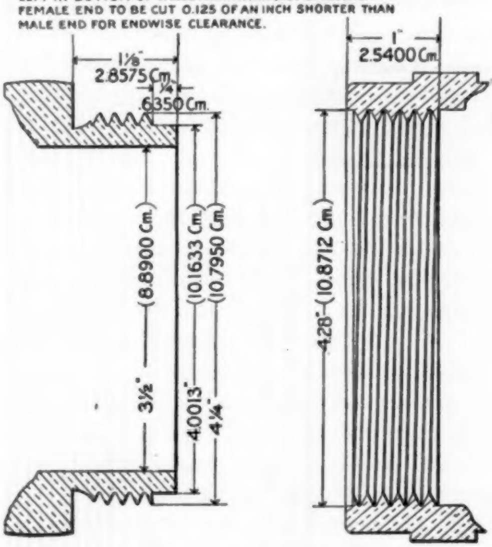
Standard couplings, thus securing a gradual and inexpensive method of standardizing the whole equipment of the city.

*e* Couplings of new hose, whenever purchased, should be the National Standard, and specifications under which new

NATIONAL STANDARD HOSE COUPLING

| | INCHES | CENTIMETERS |
|---|---|---|
| INSIDE DIAMETER OF HOSE COUPLING | 3.5000 | 8.8900 |
| BLANK END OF MALE PART | 0.2500 | 0.6350 |
| OUTSIDE DIAMETER OF THREAD FINISHED | 4.2500 | 10.7950 |
| DIAMETER OF ROOT OF THREAD | 4.0013 | 10.1633 |
| CLEARANCE BETWEEN MALE AND FEMALE THREADS | 0.0300 | 0.0762 |
| TOTAL LENGTH OF THREADED MALE END | 1.1250 | 2.8575 |

NUMBER OF THREADS PER INCH ......... 6
PATTERN OF THREAD ......... 60° V
CUT OFF AT TOP OF THREAD ......... 0.01 OF AN INCH
LEFT IN BOTTOM OF VALLEY ......... 0.01 OF AN INCH
FEMALE END TO BE CUT 0.125 OF AN INCH SHORTER THAN MALE END FOR ENDWISE CLEARANCE.




3½" SIZE (8.8900 Cm)

FIG. 3 DETAILS OF 3½-IN. COUPLING

hose is purchased should always include a clause to this effect.

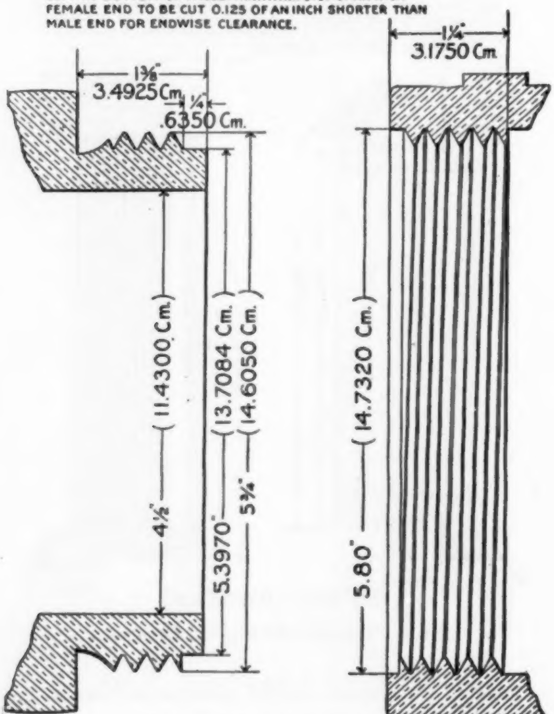
*f* Until all hose on hand has been provided with standard couplings or has been converted as suggested on the preceding page, a sufficient number of adapters should be carried on each hose wagon, so that the unconverted hose can be coupled up with the standard outlets of hydrants or fire engines.

*g* In view of the fact that 3-in. hose is coming into more general use, it is deemed advisable that such hose should be fitted with 2½-in. couplings having threads which conform to those on 2½-in hose already in use.

NATIONAL STANDARD HOSE COUPLING

| | | |
|---|---|---|
| INSIDE DIAMETER OF HOSE COUPLING | 4.5000 | 11.4300 |
| BLANK END OF MALE PART | 0.2500 | 0.6350 |
| OUTSIDE DIAMETER OF THREAD FINISHED | 5.7500 | 14.6050 |
| DIAMETER OF ROOT OF THREAD | 5.3970 | 13.7084 |
| CLEARANCE BETWEEN MALE AND FEMALE THREADS | 0.0500 | 0.1270 |
| TOTAL LENGTH OF THREADED MALE END | 1.3750 | 3.4925 |

NUMBER OF THREADS PER INCH ......... 4
PATTERN OF THREAD ......... 60° V
CUT OFF AT TOP OF THREAD ......... 0.01 OF AN INCH
LEFT IN BOTTOM OF VALLEY ......... 0.01 OF AN INCH
FEMALE END TO BE CUT 0.125 OF AN INCH SHORTER THAN MALE END FOR ENDWISE CLEARANCE.




4½" SIZE (11.4300 Cm.)

FIG. 4 DETAILS OF 4½-IN. COUPLING

It is believed that the total expense involved in a complete change from existing to standard conditions will not exceed the cost of the operations described in *b*, *c* and *f*, and that no further steps will be

needed in any city save to order all new equipment of every description to be supplied with National Standard threads.

It is of course clear that a similar line of action, as noted in *a, b, c, e* and *f,* should be followed in the case of the couplings and hydrant outlets pertaining to the suction hose of the engines. While the cost will be greater per outlet, the outlets to be thus equipped will be much less in number than for the 2½-in. connections.

While the extremes of diameter in couplings as herein indicated appeal to this committee as being conservatively reliable for the treatment recommended, many instances of adaption have been recorded wherein the deviations treated range as low as 3 in. and as high as $3\,{}^{3}/_{32}$ in. with satisfactory results in service, thus strongly emphasizing the value of the "National Standard" as an intermediary or accommodation thread coupling of wide adaptability.

It is recommended that the Higbee style of cutting the thread be adopted hereafter in order to facilitate speed in coupling up and in avoiding crossing.

These specifications, covering the essential features for hose couplings and hydrant fittings for public fire service, have been agreed upon in joint conference with accredited representatives of a number of organizations and associations interested in or controlling this class of work. They will be known as the National Standard, and to date have been adopted by the following associations:

American Public Works Association, American Society of Municipal Improvements, American Waterworks Association, International Association of Fire Engineers, League of American Municipalities, Minnesota States Firemen's Association, National Board of Fire Underwriters, National Fire Protection Association, National Firemen's Association, New England Waterworks Association, North Carolina State Firemen's Association, Pennsylvania Waterworks Association, Virginia State Firemen's Association.

Respectfully submitted,

| | |
|---|---|
| John R. Freeman, *Chairman* | *Sub-Committee on Fire Protection* |
| E. V. French, *Vice-Chairman* | |
| Albert Blauvelt | |
| F. M. Griswold | |
| H. F. J. Porter | |
| T. W. Ransom | |
| I. H. Woolson | |

## No. 1399

# STANDARDIZATION OF PIPE THREAD GAGES

The purpose of the Committee on Standardization of Pipe Threads has been to fix manufacturing limits for the use of the Briggs standard pipe thread gages when tapping fittings or flanges, so that pipe cut to the Briggs standard might always enter a definite number of turns. Although the Briggs standard is used almost universally for pipe threads in the United States, the method of its use for female threads has not been established, in that no determinations have ever been made of the standard depths to which hand plug gages should enter. This has resulted in much confusion in the past, inasmuch as pipe threaded to the Briggs standard is liable to vary in the number of threads it would screw into fittings tapped at different shops. This tendency is so marked that pipe fitting is handled in practically all cases by sending the flanges to the shop where the pipe is being cut, in order to be sure of satisfactory results.

This matter is conceded to be a simple one in that all it requires is an agreement among the manufacturers of fittings as to the point at which a ring should be attached to the gage, to establish, when the gage is inserted by hand, the proper depth of the thread. To this end the committee has met in conference with representatives of the manufacturers and also of the committee of the Society on International Standards for Pipe Threads. C. A. Olson, Chairman of the Manufacturers' sub-committee on Pipe Thread Gages, stated at this meeting that his committee had made a study of present practice among the various manufacturers and had adopted tentative definitions of the gages to be used, of the proposed thickness of ring gages acceptable to the manufacturers, and of the tolerances to be allowed. These he submitted as follows:

The gages shall consist of one plug and one ring gage of each size.

The plug gage shall be the Briggs standard pipe thread as adopted by the manufacturers of pipe fittings and valves, and recommended by The American Society of Mechanical Engineers in 1886. The plug is to have a flat or notch indicating the distance that the plug shall enter the ring by hand.

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

The ring gage is to be known as the American Briggs standard adopted by the Manufacturers' Standardization Committee in 1913, and recommended by The American Society of Mechanical Engineers, the Committee on International Standard for Pipe Threads, and the Pratt & Whitney Company, manufacturers of gages. The thickness of the ring is given in the following table. It shall be flush with the small end of the plug. This will locate the flat notch on the plug flush with the large side of the ring.

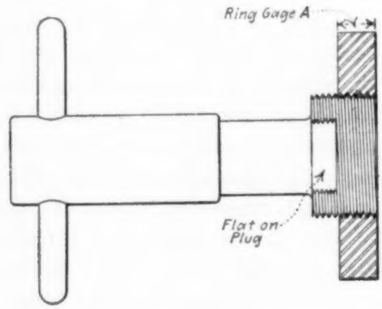



FIG. 1 VIEW OF PLUG AND RING GAGE

| PIPE SIZE | ADOPTED THICKNESS OF RING GAGE A |
|---|---|
| 1/8 | 0.1801 |
| 1/4 | 0.200 |
| 3/8 | 0.240 |
| 1/2 | 0.320 |
| 3/4 | 0.339 |
| 1 | 0.400 |
| 1 1/4 | 0.420 |
| 1 1/2 | 0.420 |
| 2 | 0.436 |
| 2 1/2 | 0.682 |
| 3 | 0.766 |
| 3 1/2 | 0.821 |
| 4 | 0.844 |
| 4 1/2 | 0.875 |
| 5 | 0.937 |
| 6 | 0.958 |
| 7 | 1.000 |
| 8 | 1.063 |
| 9 | 1.130 |
| 10 | 1.210 |
| 12 | 1.360 |
| 14 | 1.562 |
| 15 | 1.687 |
| 16 | 1.812 |
| 18 | 2.000 |
| 20 | 2.125 |
| 22 | 2.250 |
| 24 | 2.375 |

REMARKS

**The table indicates the dimensions of the ring gage *A* shown in Fig. 1, which are the figures adopted by the Manufacturers' Standardization Committee.**

**In the use of the plug gage shown in Fig. 1, the notch on the plug is to gage, and one thread large or one thread small must be the inspection limits.**

**In the use of the ring gage, male threads are to gage when flush with small end of ring, and one thread large or one thread small must be inspection limits.**

After consideration of the matter by the entire committee, it was voted that the table of sizes and tolerances submitted be approved as above, by the joint committee, and that there be deposited with the Bureau of Standards at Washington, D. C., a set of gages to be known as the American Briggs Standard for Pipe Threads, the expense of the manufacture of these gages to be provided for at a conference between the Manufacturers' Committee, the Bureau of Standards, and the Pratt & Whitney Company.

Respectfully submitted,

| | |
|---|---|
| E. D. MEIER, *Chairman*<br>GEORGE M. BOND<br>A. M. HOUSER<br>J. C. MELOON<br>CHARLES A. OLSON<br>PETER C. PATTERSON<br>HENRY G. STOTT | *Committee on Standardization of Pipe Thread Gages* |

No. 1400

# NOTES ON THE FURTHER OPERATION OF LARGE BOILERS OF THE DETROIT EDISON COMPANY

By J. W. Parker, Detroit, Mich.

Junior Member of the Society

At the December 1911 meeting of the Society, Dr. D. S. Jacobus presented a paper[1] giving the results of the first performance tests on the 2365-h.p. Stirling boilers at the Delray generating plant of the Detroit Edison Company. At the time of his tests three of these boilers were in service, one having been run about 18 months and the others nine months. Since that time six more of the type have been installed at the rate of two a year, the last two in the autumn of 1913. It is the object of this paper to present some report of the everyday experience in operating all these boilers.

2 *Reliability.* That a boiler unit can be made so reliable that it may in this respect be classed with the turbo-generator, has been the experience with the nine boilers installed at Detroit. Table 1 shows the performance of six of these units during October, November and December of 1912. This period was selected as being the time when reliability was essential on account of the plant load conditions. The load curve of the plant shown in Fig. 2 is typical. At no time was it compulsory to put a boiler out of commission; at no time was any boiler out of commission during the peak load of the day, from four o'clock to six o'clock in the afternoon. Of the 215 hours the three Taylor stoker-fired boilers were taken out of commission, but 89 hours occurred during the five peak days of the week, Monday to Friday. This time was employed in cleaning furnaces, repairing stokers and inspecting tubes externally. The boilers themselves proved to be 100 per cent reliable.

3 *Possible Cause of Trouble.* The causes which might take boilers out of commission at times when they are needed for service, are being eliminated. There are three sources from which to expect trouble:

[1]Trans. Am.Soc.M.E., vol. 33, p. 565.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

(*a*) the boiler proper, (*b*) the furnace brickwork, and (*c*) the stoker.

4 (*a*) The boiler itself need be no hazard. Fittings and flanged joints, if properly watched and followed up, do not cause shutdowns. The same is true of the tubes, which are seamless. From October 15, 1912, to November 1, 1913, there have been but two tubes replaced in seven boilers whose average age is two years. One front tube was taken out after the discovery of a small bag next to the fire, which was found during a periodical external inspection of the front tubes during furnace repairs. The other tube was in the back row and was

TABLE 1 SHOWING TIME BOILER UNITS WERE OUT OF COMMISSION

| | Boilers Nos. 25, 27 and 33 Taylor Stoker | | Boilers Nos. 26, 28 and 30 Roney Stoker | |
|---|---|---|---|---|
| | Hours | Per Cent | Hours | Per Cent |
| Total Number of Boiler-Hours (3 months) | 5160 | .... | 6624 | .... |
| Number of Boiler-Hours out of Service (cleaning) | 298 | .... | 323 | .... |
| Total Number of Boiler-Hours available for Service | 4862 | .... | 6301 | .... |
| Total Steaming Hours | 3178 | 65.4 | 3594 | 57.0 |
| Total Banking Hours | 1469 | 30.2 | 1764 | 28.0 |
| Out of Service at Night | 26 | 0.5 | .... | .... |
| Out of Service at Week End | 100 | 2.1 | 219 | 3.5 |
| Out of Service for Work-day Repairs | *89 | 1.8 | †724 | 11.5 |
| Total | 4862 | 100.0 | 6301 | 100.0 |

* Cleaning Furnace, Inspecting Tubes and Repairing Stoker, 89 hours.

| | Hours |
|---|---|
| † Repairing Stoker | 537 |
| Repairing Furnace Wall | 173 |
| Packing Blowoff Valves | 14 |
| Total | 724 |

spoiled by a mishandled turbine tube cleaner. Its condition was discovered while the soot was being blown from the heating surfaces by hand blowers. However, the leak was very slight and the boiler was not cut out of service until night.

5 The plant is using about 8 per cent of make-up water, and although Detroit River water at Delray is not considered bad for boilers, nevertheless, a sulphate scale about 1/16 in. thick forms in the hot front tubes. These front surfaces are worked exceedingly hard and scale of even 1/16 in. thickness may make trouble. A soft piece of mud, lodging on top of the hard scale is likely to cause a small bag.

This trouble can occur only in the lower bends of the first row tubes, where loose dirt sometimes lodges.

6 Tube trouble will be practically eliminated by the use of pure water. The make-up water is now river water, treated with enough barium hydrate to neutralize the sulphate content. The percentage of make-up can be reduced to between 2 and 3 per cent by avoiding

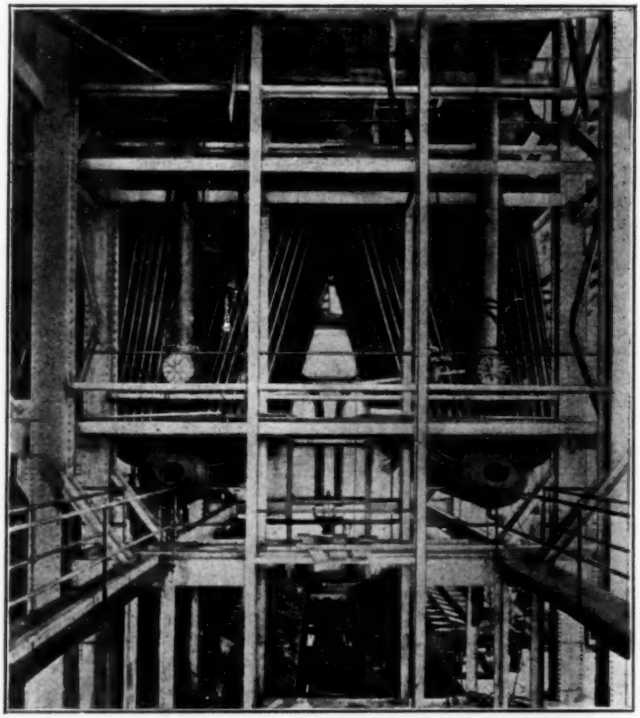


Fig. 1 General View of 2365-h.p. Boiler and Stoker before being bricked up

the necessity of blowing down boilers. It is seriously contemplated to use distilled water for make-up. There seems to be a lack of evidence that distilled water is of itself corrosive. The cost of distilling would be almost negligible, since the heat required to evaporate the raw water would be kept in the system by using main turbine condensate for cooling water in the distillation process. To safeguard against any possible boiler corrosion, enough soft scale forming ma-

terial would be introduced into the feedwater to cover the heating surfaces with a thin protecting scale.

7 (*b*) Furnace brickwork is with the underfeed stoker reduced to a minimum as evidenced by Fig. 3. It consists of the four furnace walls and nothing more. Much work has been done to make these

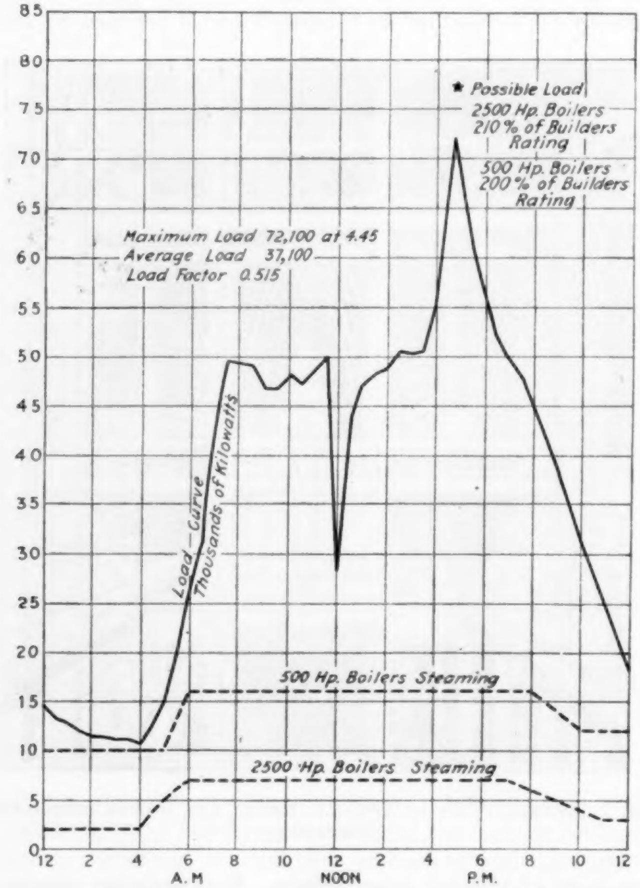



Fig. 2 Peak Load Curve, December 6, 1912

walls stand up. It has been necessitated on the ground of maintenance cost only, however, since if certain sections of the furnace are rebuilt once a year, there is no chance for sudden trouble which would cause a shutdown. There has been no such sudden trouble.

8 (*c*) The stoker remains to be considered. The fallibility of the

stoker is not great. Repairs ordinarily made are the replacement of tuyeres, dump plates and the like which have burned out. The work, mentioned in later paragraphs of this paper, which has been done to decrease stoker maintenance costs has, as in the case of brickwork, made it unlikely that the boiler need ever be taken out of commission for this cause, except when the plant load conditions permit it.

9 There have been cases of sudden stoker breakdown, sometimes

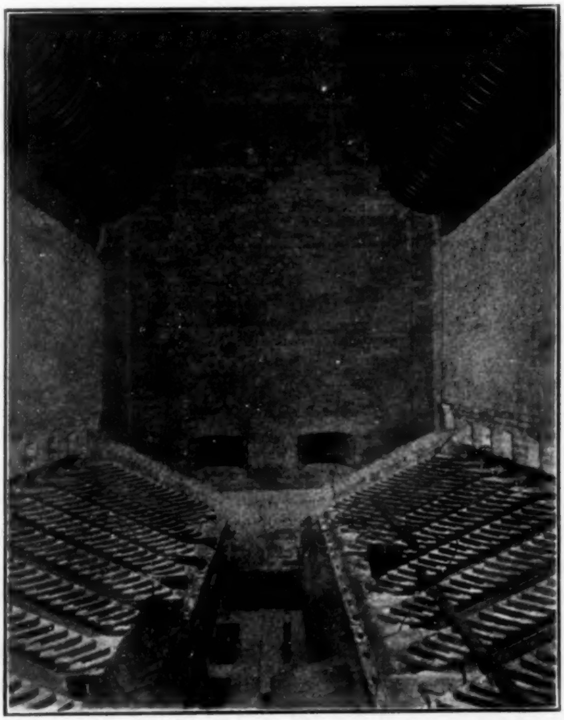


Fig. 3 Inside View of Furnace with Taylor Stoker

occasioned by the introduction of chunks of iron or wood into the stoker rams, along with the coal. Also, there have occurred once or twice, serious burn-outs of tuyeres and other grate surfaces extending over an area of 5 or 6 sq. ft. In these instances, the fire can be, and has been, banked on the one side of the furnace where the trouble exists and the other half of the furnace operated to get half capacity from the boiler. This one-side firing is entirely practicable.

10 To sum up, the experience has been this: The boilers were ready for service 95 per cent of the time considered; they were ready 98 per cent of the five full load days of the week; and ready for service 100 per cent of the time during the peak load periods. It is to be assumed that the boilers must be down for ten days or two weeks at nine or ten month intervals for boiler cleaning and a general overhauling of stokers and brickwork.

11 *Recent Design.* On the above considerations, the Detroit Edison Company is now building a power plant to contain six 20,000-kw. turbines served by 12 of these 2365-h.p. boilers, which is two boilers to one turbine, with no spares, or 10,000 kw. per boiler. Although the chances of losing the capacity of any boiler by a forced shutdown are remote, they have nevertheless been considered. At normal full

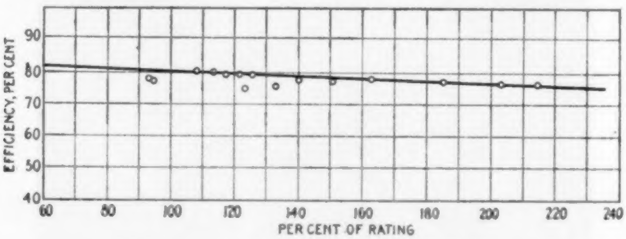



FIG. 4 CURVE SHOWING OVER-ALL EFFICIENCY OF BOILER AND FURNACE AT DIFFERENT RATINGS—TAYLOR-STOKER FIRED UNITS

load on a given turbine unit, the two boilers will operate at approximately 191 per cent of the builder's rating based on 10 sq. ft. of heating surface per boiler horsepower. If, with six boilers running at this rating, one boiler should go completely out of commission, the other five would have to carry the entire load of 60,000 kw. and thus operate at 235 per cent of rating, which is perfectly possible. As a matter of fact, the settings and auxiliaries are being designed to allow of continuous operation at 255 per cent of rating, which would enable three boilers to take the full load of four, i. e. 40,000 kw. Very recently, one of the Delray boilers was isolated from the rest of the plant with a 15,000-kw., seven-stage Curtis vertical turbine, and over 11,000 kw. was carried for an hour without difficulty.

12 *Flexibility.* The over-all boiler and furnace efficiency of the Taylor stoker-fired unit tested by Dr. Jacobus is shown by Fig. 4. The curve is a straight line throughout the range tested, having a gradual slope from 80 per cent efficiency at 93 per cent of rating, to 76 per cent efficiency at 214 per cent of rating. It is economical,

therefore, to run as many boilers as possible at about 90 per cent of rating when the plant load is light, and then carry the peak of the load by increasing the rating on the boilers. This is our present practice. In this way, from morning till night there need be no fires banked or broken out of bank, and the firemen can bend their energies instead to manipulating their fires to the best advantage.

13 This flexibility is at no time more convenient than in summer when provision must be made for a sudden peak load due to a thunder-

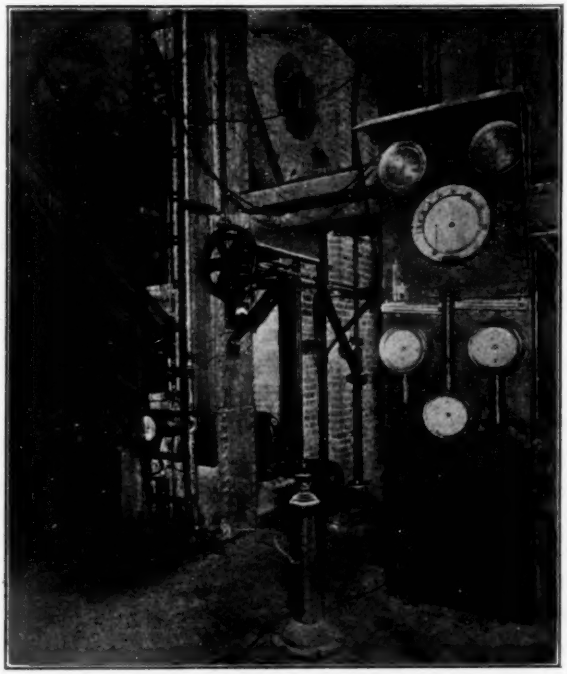


Fig. 5 Gage Board and Control Point for One Boiler Unit

storm. At Delray, in the summer of 1914 the average day load will be about 63, 000 kw., while provision must be made for a storm load of about 82,000 kw., a 30 per cent increase. Boilers ordinarily running at 100 per cent or 125 per cent of rating (a very economical point) will take a 30 per cent increase in load with very little effort. No banked fires will be carried during the daytime.

14 *Control Methods.* A boiler of such output assumes enough of importance and individuality, so to speak, in a 120,000-kw. plant to

gain for itself unusual attention from the man who is firing it. One fireman fires two units. The control of each unit is brought directly under his hand in every way possible, so that a minimum of time will be wasted in mechanical manipulation. Fig. 5 is a view of the gage board and control point for one boiler. A water tender stationed on a gallery at the top drum level feeds the boiler but the fireman does everything else in the way of operating. The boiler dampers are shifted at the control point, and the throttles of the blower turbines are extended through the floor, the valve stand being seen in the view, near the gage board. The stokers are motor driven and one of the controllers is located on a steel column 6 ft. away from the gage board; the other could have been brought around also, but it was thought unnecessary, since a man must walk around to the stoker on the distant side to watch it and to study his fire. The control point of the opposite boiler is located just across the aisle from the first.

15 The plan is for one man to operate two stokers, and in addition, to have a head fireman in charge of from six to eight units, whose duty it is to oversee all the fires, and go to the assistance of any fireman who needs help. On the gage board are mounted steam gages showing pressure at the superheater inlet and superheater outlet and draft gages showing air pressure under the fire, draft at the damper, and draft at the top of the combustion chamber. There is also on this board the record dial of a $CO_2$ meter. The meter itself is mounted on a gallery 15 ft. above the boiler room floor in a position central of four boilers. Four samples of gas are drawn from one furnace, automatically mixed, and the resulting analysis is recorded where the fireman can watch it.

16 *Fireroom Personnel.* The whole idea is to employ the most expert firemen it is possible to develop, and give each man control of the burning of a very large amount of coal. It is economical to employ a fine type of man and give him an expert's pay. The present first-class fireman's pay is 40 cents an hour and he is well treated as to vacation and sick leave. These men are given the benefit of whatever boiler testing is done, and a force of firemen is being built up that can obtain remarkable results with their fires. They are acquiring an intelligent working understanding of the combustion of coal. At the same time the unit cost of firing will be unusually low. Table 2 is a schedule of the labor necessary to handle 12 boilers of a six-turbine plant with no economizers installed.

17 The number of men allowed for such a plant is ample as shown

by long experience at Delray. The cost per day is 55 per cent, as much as the firing cost in a neighboring manufacturer's plant equipped with twenty-four 278-h.p. Babcock & Wilcox boilers fired by overfeed stokers and equipped with economizers. In this plant it was found that maximum boiler and furnace efficiency was obtained while running the boilers at 95 per cent of rating, and, with a load factor of nearly 100 per cent, it was considered economical to operate at that rating. In order to make a comparison of the cost of firing large units and

TABLE 2 LABOR COST OF FIRING A TWELVE BOILER PLANT

| | | | |
|---|---|---|---|
| Maximum load | 120,000 kw. | | 12 boilers at 191 per cent |
| Minimum load | 20,000 kw. | | 4 boilers at 96 per cent |
| Monthly load factor (November) | | | 46 per cent |
| Operators employed— | | | |
| Morning shift | 6.30- 2.30 | 2 head firemen at 45 cents | $7.20 |
| | | 6 firemen at 40 cents | 19.20 |
| | | 2 watertenders at 35 cents | 5.60 |
| Afternoon shift | 2.30-10-30 | 2 head firemen | 7.20 |
| | | 6 firemen | 19.20 |
| | | 2 watertenders | 5.60 |
| Night shift | 10.30- 6.30 | 6 firemen | 19.20 |
| | | 1 watertender | 2.80 |
| | | | $86.00 |
| Boiler room foremen | | | 15.00 |
| Total cost per day | | | $101.00 |
| Cost of firing boilers — cents per kw.-hr | | | 0.0076 |
| Actual cost in a neighboring manufacturing plant's boiler room to generate an equal amount of energy but at 100 per cent load factor | | | $169.00 |
| Estimated cost in neighboring manufacturing plant's boiler room if load factor is 46 per cent (allowing 191 per cent maximum boiler rating) | | | $182.00 |
| Scale of pay—watch foreman | | | 31½ cents per hour |
| firemen | | | 25 cents per hour |

small units, it has been assumed that the small boilers could be operated over the same range as the large ones. At the same time it has been assumed that both plants operate with the same load factor. The comparison is greatly in favor of the large unit, notwithstanding the fact that the scale of pay on these boilers is much higher than that on the 278-h.p. boilers. It is to be noted that the load factors are for monthly periods.

18 *Pilot Steam Gage and Indicators.* At the end of each firing aisle is mounted a large pilot steam gage. The dial is graduated in divisions of 1½ in. on its circumference but with no figures, and each

scale division registers 1 lb. per sq. in. It is found and marked at just what point on this sensitive gage the boiler safety valves will lift and the steam pressure is carried accordingly. On the same gage board with the pilot gage are a clock and the dial of a load indicator, operated from the switchboard gallery, indicating the load on the entire plant in kilowatts. In handling his fires, the fireman is guided in the first place, of course, by steam requirements as shown by the pilot gage and a steam flow meter. The air pressure under the fire and the stoker speed are controlled by hand.

19 *Variable Stoker Speed.* As before mentioned, the stokers are driven by motors. These are direct-current interpole machines with a speed control by field resistance of four to one, and with 18 running points on the controllers. Indicating dial tachometers are being installed on the stoker shafts, so that the stoker speed for a given load may be watched carefully. An immense advantage is gained by being able to run the stoker at any given constant speed. The amount of power required to drive the stokers is extremely variable, and it is difficult to make a steam stoker engine govern under these conditions with a controlled range of speed variation of four to one.

20 As to furnace conditions, the firemen must judge by the $CO_2$ recorder, by the amount of air pressure necessary for any given boiler load, and by no means least of all by the color of the gases as they tumble over the first baffle and enter the top of the superheater pass, Fig. 6. A reflecting observation device for this is at present being tried out, which allows the fireman to look into this boiler pass from a position near his gage board, instead of, as now, being dependent for this valuable indication on the water tender's or foreman's observation.

21 *Possibilities of Automatic Control.* It is a question not yet entirely settled whether it would or would not be better to forsake hand control of these boilers for automatic control. It has been suggested that feedwater regulators be used and that some automatic steam pressure governed apparatus control the air pressure under the fire and the stoker speed. That is not necessitated by labor cost; neither by the need of steam pressure regulation, as is evinced by the reproduction of the steam pressure chart in Fig. 7; this chart covers January 7, 1913, a day with wide variation of load. Our prejudice, if not our final judgment, is against automatic control.

22 As for feeding the boiler, a skilful water tender learns not to touch his feed valve except at rare intervals and when the load on the plant becomes steady, the steam flow is continuous also. One auto-

matic feedwater regulator was experimented with. It was not a success, due in most part to a peculiarity of this type of boiler. On account of the rapid circulation, the water in the top middle drum stands at times as much as 2 ft., or more, higher than in the two outside drums, although the three are connected with water circulating tubes as well as by steam circulators and the main boiler tubes. The water columns are piped to the center drum, there being often no indication of water whatever in the outside drums.

23 The water level as shown in the glass under the above conditions, naturally, is variable, being a function of the rate of steaming. This being the case, it is easy for a feedwater regulator, automatically attempting to hold a constant level in the center drum, to get into difficulties whenever the demand for steam varies sharply, as it will, when the plant load is changing at noon or during the peak load of the day. When the steam demand decreases slightly, the water level may fall possibly an inch, being very sensitive. If the regulator immediately opens and starts feeding more heavily, the circulation will be still further decreased, and then the water level will fall lower and the regulator feed still more heavily. The steam output of the boiler will decrease rapidly with this, until the regulator begins to close again. The steam output will now begin to increase and the water level rise also and presently close the regulator entirely; a typical "hunting" action is thus set up. Some type of regulator can undoubtedly be worked out so as to handle these boilers; but it is questionable whether it would be worth while, when one water tender can tend water for a 60,000 kw. load.

24 *Automatic Control of Fuel and Air.* There remains the matter of handling coal and air feeds. In a small plant owned by the Detroit Edison Company, eight stokers are controlled automatically, both air and coal feed being independent of the fireman. The latter is not allowed to touch the stoker adjustments except to change the steam pressure one way or the other. This is going to the extreme, but it is not an uncommon method of firing underfeed stokers. It certainly does not tend to develop intelligent firemen nor obtain very satisfactory furnace efficiencies.

25 It is not desirable to substitute an automatic air and stoker control if a better $CO_2$ chart cannot be shown thereby. The danger is that the fireman will rely on the automatic device, rather than study conditions and control his fire intelligently. With the present hand control, excellent furnace conditions are being obtained and held.

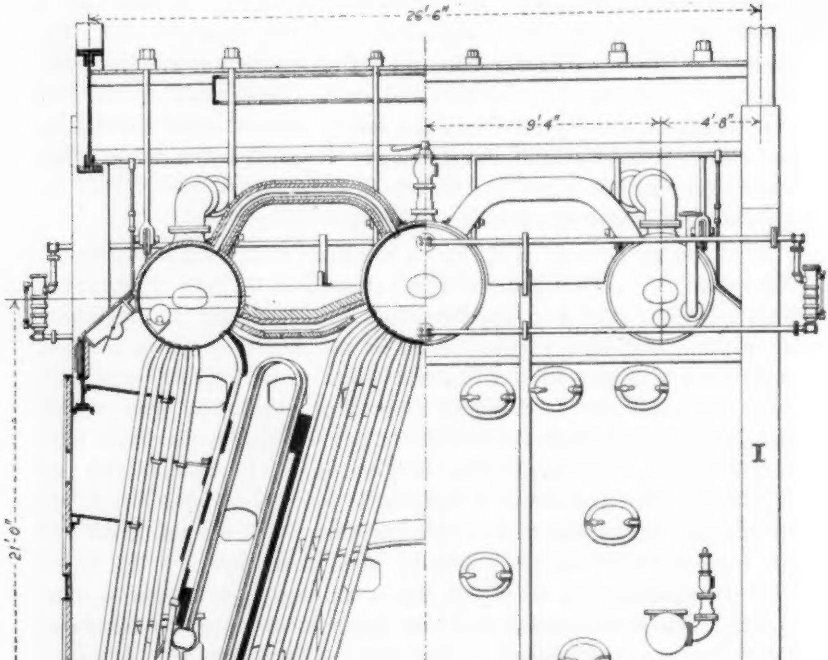



FIG. 6 END SECTION OF BOILER SHOWING ARRANGEMENT OF GAS PASSES

Observation of the furnace gases, as they enter the superheater pass from the top of the combustion chamber, shows that the combustion of volatile gases is entirely complete and that the operation is consequently smokeless. At the same time, the $CO_2$ charts show remarkably good results, 15 per cent of $CO_2$ being very common, the average being about 13.5 to 14 per cent. Repeated analyses made with an Orsat apparatus check these recording machines and at the same time discover no more than from a trace to 2/10 of a per cent of carbon monoxide.

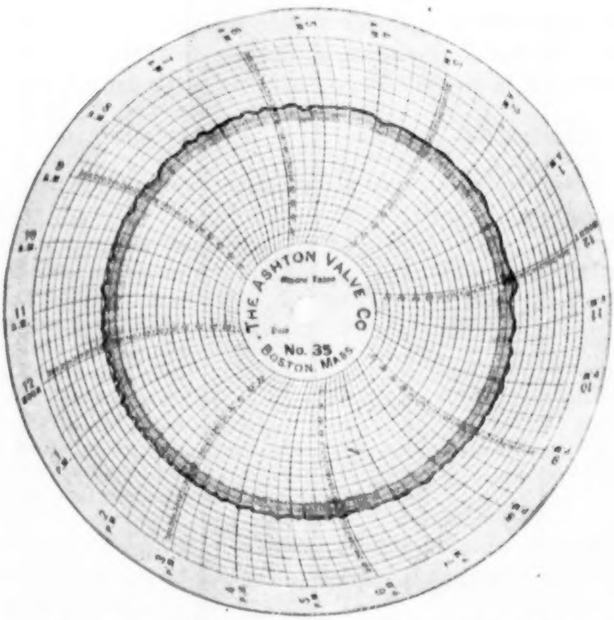



FIG. 7 STEAM PRESSURE CHART, JANUARY 7, 1913

26 Good conditions are most often disturbed by the periodic dumping of the ash and refuse from the stokers. A continuous dumping device, a revolving grate, is now being tried out, which promises to be very successful. It will undoubtedly go far toward enabling the fireman to produce service results as good as those shown during tests.

27 *Maintenance Costs.* A careful study was made for four months in 1911 and 1912 of the maintenance costs of the boiler units fired by underfeed stokers and the figures are presented in Table 3. Since the period selected includes one job of general overhauling of a stoker

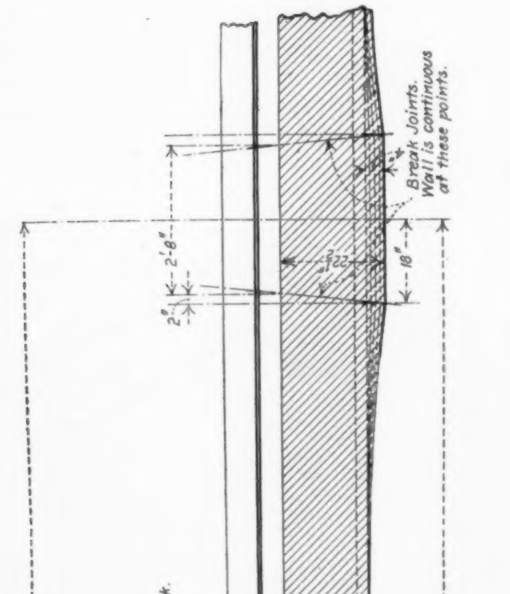



FIG. 8 DETAILS OF VERTICAL ARCH CONSTRUCTION OF FURNACE WALLS

and setting, it is conservative to estimate the maintenance cost for a whole year from these four months. This allows for the overhauling of boilers at intervals of eight months on the average, which is the present practice in the Delray plant. In this way the annual cost per kw-hr. generated has been estimated, which includes maintenance of stokers, settings and boilers.

28 This maintenance has, as a matter of fact, decreased since the time referred to, because of improvements worked out on the stoker dumps. The dumping plates have now been made with removable tops, so that only the part burned need be removed and the web of the casting may be left in place. By a redesign of the shape of these dumping sections only one pattern is made use of, and the labor cost of replacing burned parts has been greatly reduced at the same time.

TABLE 3 MAINTENANCE COSTS OF TWO 2365 H. P. BOILER UNITS WITH TAYLOR STOKERS

INCLUDING REPAIRS TO BOILERS, STOKERS AND BRICKWORK

| | Boiler No. 25 | Boiler No. 27 |
|---|---|---|
| December 1911—Hours steaming | 442 | 447 |
| Cost of material and labor | $116.00 | $112.60 |
| January 1912—Hours steaming | 483 | 490 |
| Cost of material and labor | $84.86 | $2.88 |
| February 1912—Hours steaming | 464 | 462 |
| Cost of material and labor | $63.35 | $77.74 |
| March 1912—Hours steaming | 424 | 182 |
| Cost of material and labor | $1.71 | $253.02* |
| Cost per kw.-hr. in 12 boiler installation, with two boilers for one 20,000-kw. turbine (yearly load factor 36 per cent) | | 0.0034 cents |

*Covers general overhauling of stoker and setting.

In 1912, one stoker was completely rebuilt in order to experiment with various changes of design, especially of dumps, which have required most frequent renewal.

29 *Setting Repairs.* The settings of these boilers consist, as was stated before, simply of four walls without arches or bridge walls (see Fig. 3). The only brick repairs necessary thus far with this setting have been the annual rebuilding of part of both front walls (area 150 to 175 sq. ft.) extending from the stoker to the mud drum of the boilers. The end walls give no trouble when carefully bonded to the facebrick of the setting. As an experiment, the front walls of one unit are to be built of firebrick shapes, supported by a cast-iron or steel supporting frame. This construction will involve a sheet steel casing to make the setting air-tight, and will be less like a wall, and

more like a steel casing protected from the fire by firebrick which are not self-supporting. In the last two units installed this autumn the walls are built with the surfaces slightly concaved on the fire side. The plan section is an arch, as shown by the detail drawing reproduced in Fig. 8.

30 Since each individual firebrick in a wall exposed to the fire on one side tends to expand permanently on the hot end, the tendency also is for each brick to pull in towards the fire. Bonding and anchor bolts ordinarily prevent this, but at some points, the influence of bonds and local anchoring is not enough to prevent large areas of the wall from falling toward the fire. The object of the "vertical arch" design referred to above is to wedge each individual brick into the wall by

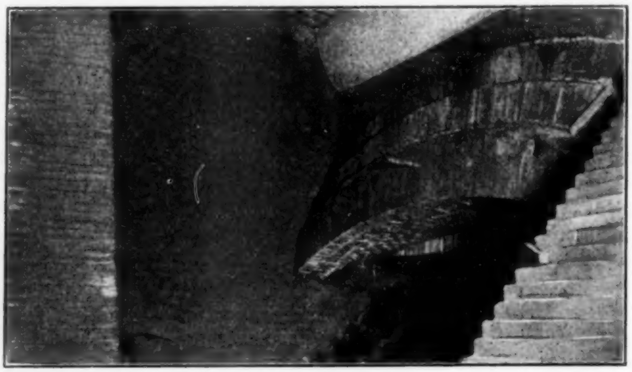


FIG. 9 PHOTOGRAPH OF ARCH FOR RONEY STOKER

an arch construction and thus prevent the walls from bulging. The results, of course, remain to be seen.

31 *Roney Stoker Settings.* The most educative experience with brickwork has been had with the Roney stoker settings and their furnace arches. Starting with two sprung arches under each mud drum with a central wall intersecting the bridge wall at right angles, the design ended up with a single continuous "flat arch," so called, under the mud drum, built of fire brick shapes hung from cast-iron rail beams (see Figs. 9, 10 and 11). For these shapes a special brick had to be made of an exceedingly loose texture obtained by mixing sawdust with the clay before molding. Upon burning in the kiln, this sawdust burned out, leaving the brick very porous. It was found that a harder, closer grained brick invariably spawled under the repeated heating and cooling to which it was subjected.

32 *Limitations of Boilers as at Present Installed—Furnace Height.* As at present installed, these boilers present certain limitations to being driven at any considerably higher per cent of rating than that already obtained. First, in burning West Virginia, long-flaming bituminous coal, it is probable that at, say, 275 per cent of rating and perhaps somewhat lower than that, the flames will reach the top of the combustion chamber, which is twenty-eight feet high. As soon as uncombined combustible gases get over into the superheater pass, the over-all efficiency of the unit will drop, for although secondary combustion will take place, nevertheless some unburned volatile matter must escape. Smoking will begin immediately after the secondary combustion becomes very considerable.

33 If the boiler in future installations is set much higher another difficulty will arise, although this is not so serious. Either more stack draft must be provided, or a pressure will be developed at the top of the combustion chamber, and the brick setting at this point must be built gas-tight. This phenomenon is readily explainable. Inside the combustion chamber is a column of gas like the column of gas in a stack. This stack is only 26 ft. high between draft gages in the present settings, but the gas column is at an average temperature of about 2300 deg. fahr. Figured statically this would create a draft of 0.3 in. of water at its base, providing the pressure at its top is atmospheric. However, if the maximum stack draft obtainable is reached at 250 per cent of rating with the present breeching and stack arrangement, the draft over the fire might become at 275 per cent of rating, only 0.1 in. As figured, this gas pressure over the fire must be less than at the top of the chamber by 0.3 in., neglecting friction. The pressure at the top would, therefore, become 0.2 in. above atmospheric. Pressure at this point is frequently developed in practice with the present settings.

34 *Internal Losses of Pressure.* Another limitation is the drop in pressure through the superheater, the automatic check valves and stop valve of the boiler. At 210 per cent of rating on one boiler, the drop in pressure through the superheater is 21 lb. which includes the pressure drop through the automatic check valves, but not that through the main stop valve. At 255 per cent of rating it would be considerably more. Since the pressure on the steam mains must remain constant, a pressure must be carried on the saturated steam drums of the boiler of 21 lb. more than at the superheater outlet, when running at 210 per cent of rating.

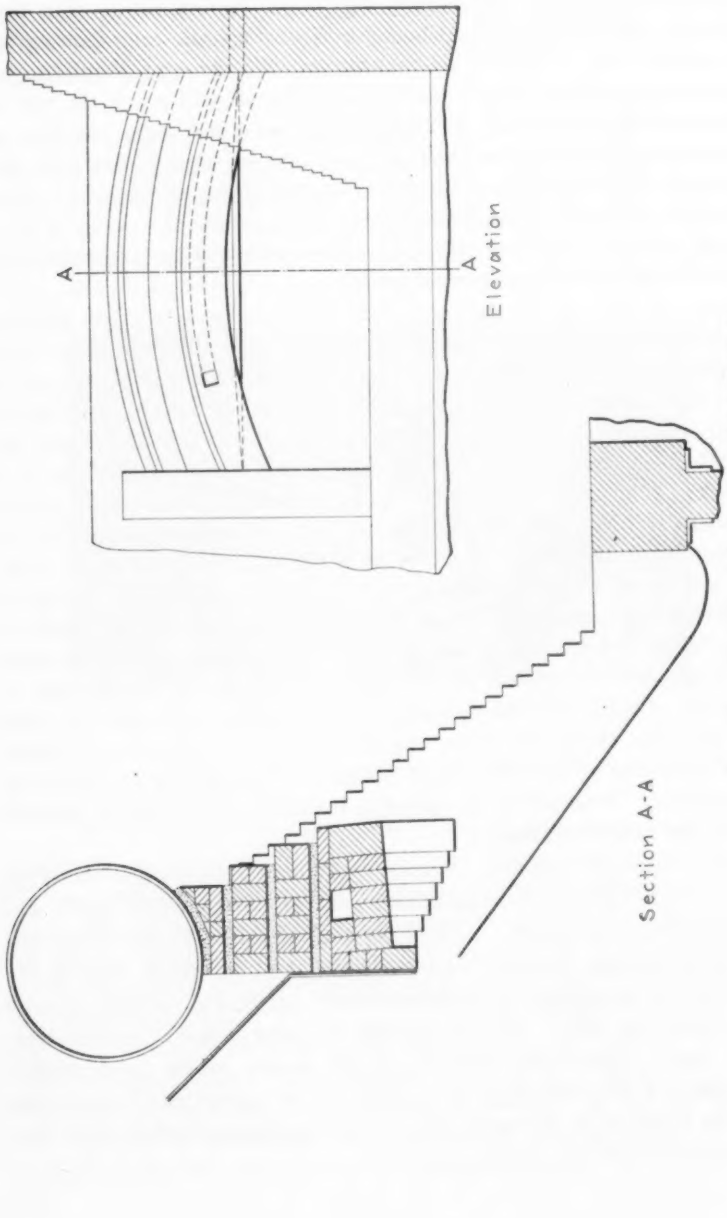



FIG. 10 DRAWING OF ARCH FOR RONEY STOKER

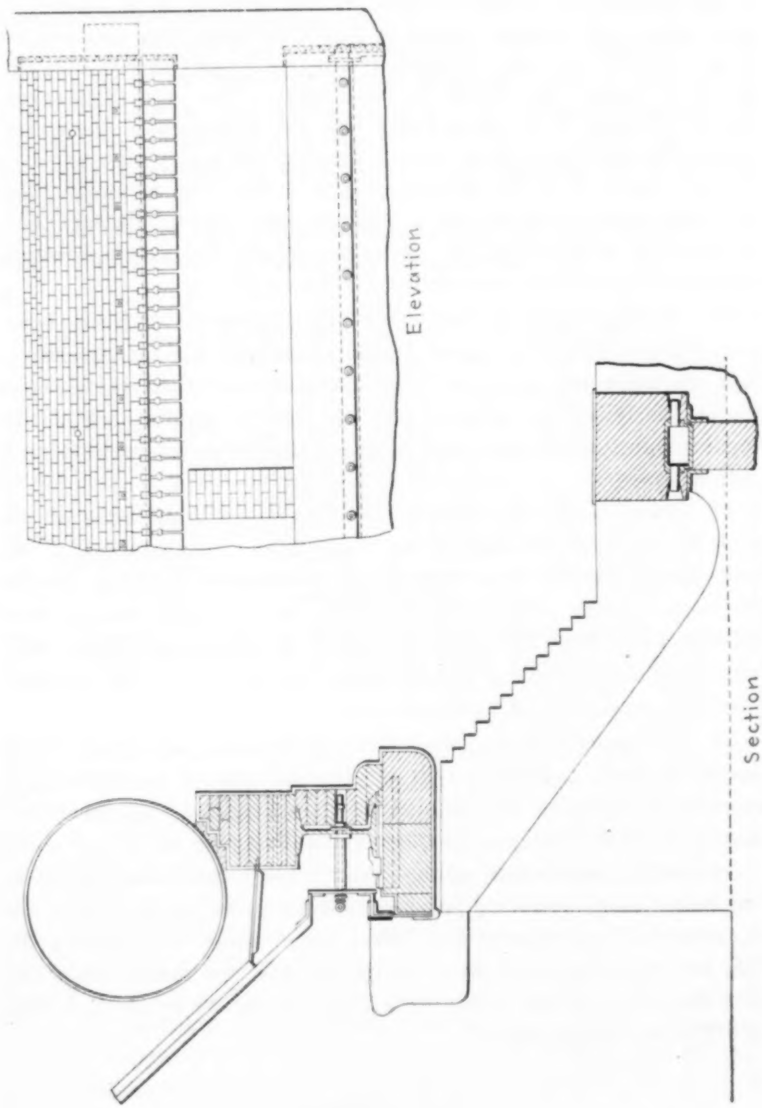



FIG. 11 DRAWING OF FLAT ARCH FOR RONEY STOKER

35 The drop through the stop valve is preventable. There were at first installed on several of these boilers, however, Hopkinson-Ferranti valves with throats reduced to 5 in. from a full pipe diameter of 10 in. At 210 per cent of rating the figured steam velocity through this 5 in. throat was 47,000 ft. per minute. The valve discharged into a Y-fitting in the main steam line and consequently, the steam passing through the valve at lowered pressure and a very high velocity, did not regain all of the pressure lost as it should in a venturi tube of correct form. The net loss in pressure after passing the valve was at times as much as 12 lb. Straight-opening Hopkinson-Ferranti valves have since been substituted for the venturi type.

36 The experience at Delray with very high steam velocities, however, has proved that in mains designed especially for high velocities, such practice is very good, the difficulties being more than compensated for by the reliability, reduced cost and ease of maintenance of the smaller diameters of mains and fittings. It is a subject worth a great deal of study.

37 *Effect on Tubes.* As for the effect on the front tubes of the type W boiler, of running at very high rates of evaporation, it has been found that the tubes have shown no evidence of injury due to the hard driving. One thing is certain, however, and that is these tubes must be kept clean. As stated in an earlier paragraph, scale which ordinarily would give no trouble, has possibilities for mischief under the conditions of harder driving.

38 The general conclusions arrived at from the experience had in operating these boilers, is that large units present possibilities of economy of operation and simplicity of power plant design, which are greatly in advance of present steam generating practice.

39 In the preparation of the data for this paper, credit is due to the boiler room operating engineer and his force, as well as to the experimental engineers of the Delray plant, whose work during the last two years has included so much fuel and combustion investigation that they are now known throughout the service by the nickname of "The Gas Department."

## DISCUSSION

R. H. Danforth (written). The results of operation of this set of boilers, as reported by Mr. Parker, are no less startling than were

the tests of the first three of these gigantic units, described by Dr. Jacobus in his unique paper.[1]

It is evident from Fig. 12, which shows a graphic comparison of the overall boiler efficiencies at different steaming rates, as obtained by Dr. Jacobus, with those shown in Fig. 4 of the present paper, that the extraordinary efficiencies obtained under test conditions have been not only maintained in service, but even improved by more than 1 per cent.

Of course this does not mean that boilers, like wine, improve with age. It can mean but one of two things in the present case, or per-

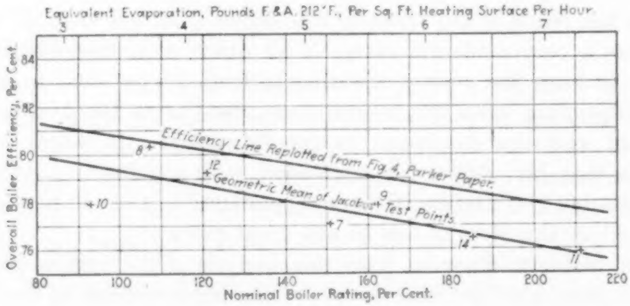



FIG. 12 COMPARATIVE CURVES SHOWING RELATION BETWEEN RESULTS REPORTED BY MR. PARKER AND THOSE OBTAINED IN DR. JACOBUS' TESTS ON THE SAME BOILERS

haps a combination of the two; namely, a reduction in the temperature of the flue gases, or an improvement in the furnace efficiency.

Mr. Parker gives no data in his paper as to the losses in the flue gases, but if we refer to test No. 8, Table 16 of Dr. Jacobus' paper (the best test there reported, showing an overall boiler efficiency of 80.28 per cent at 108 per cent rating), we find that the flue gas temperature is 493 deg. fahr. and that of the steam at 200 lb. gage is 388 deg. fahr., a difference of only 105 deg. fahr. This means that even if we could so improve this boiler as to discharge the gases to the stack at the same temperature as the boiler steam (assuming that the chimney draft thus provided would be sufficient to take away the gases), we could increase the overall efficiency of the unit by but slightly over 2 per cent.

Going again to Dr. Jacobus' paper, in the absence of sufficient

[1]Tests of Large Boilers at the Detroit Edison Company, Trans. Am. Soc. M. E., vol. 33, p. 565.

information in Mr. Parker's, we find but 1.8 per cent of the available carbon lost in the ash, a very satisfactory showing for mechanical stoker operation, and one which we believe can hardly be bettered.

In Table 5 of Dr. Jacobus' paper, for test No. 8 the CO in the flue gases was 0.02 per cent at the bottom of the last pass, 0.01 per cent at the top of the last pass, and 0.10 per cent in the flue. These discrepancies were discussed in connection with the presentation of the original paper, so we need not attempt to discover their cause. If, however, we assume even the highest of these values, 0.10 as the correct one, a completeness of combustion is indicated, which we can hardly hope to materially improve.

This test No. 8 of Dr. Jacobus' also shows a flue gas analysis which indicates a very moderate excess of air above that theoretically required to burn the coal, amounting to from 25 per cent to 40 per cent, depending upon the assumptions made as to the hydrogen in the coal and the steam-gas in the flue gases. It seems hardly credible that Mr. Parker has succeeded, with any form of stoker, in so improving the mixture of air and combustible gases in his furnace as to reduce this item of loss to any great extent.

I am unable to discover, from this analysis of the original tests on these boilers, any probable source of the improvement in their efficiency which he indicates, especially under operating conditions as against test conditions, even though the operating conditions are such ideal ones as Mr. Parker describes. I would like, therefore, to have Mr. Parker give us any data which are available, of flue temperatures, gas analyses, etc., which will indicate in what direction he has succeeded in making his gains, and obtained the marvelous operating efficiency which he reports. In so unusual a record of operation, Mr. Parker has set a mark quite as high, if not higher, than did Dr. Jacobus in their design.

Alex Dow said that the theory of the Detroit Edison management had been not merely to reduce floor space, to reduce the unit investment, and to secure a very high operating efficiency of the mechanism, but to demonstrate that in the boiler room, as in other departments, it was possible to pay larger daily or hourly rates to employees, and to get results that would warrant these rates.

Looking broadly at the proposition, the vital point had been the reduction of the number of firemen, and the education of those fewer firemen until they developed and demonstrated such interest in their work, as was evidenced by their insistence on checking their running

instruments. A very much higher class of men could be obtained if the equipment were given them to show the interesting processes which they were carrying on. The firemen at the Detroit Edison plant were beginning to comprehend that they were carrying on exact chemical processes; that they had to watch not only the draft and the appearance of the fire, but also the $CO_2$ recorder. These firemen often went to the engineering department and asked for an Orsat apparatus to check up their recorders.

Ideals were not always realized and human nature had continually to be taken into account, but Mr. Dow felt that the approximation of daily practice to theoretically perfect working was much closer in the Detroit Edison Company's boiler plant, for the present year, than in any other plant in the country.

Albert A. Cary was interested to know what the experience had been, as far as checking was concerned, with the flow meters used in the Detroit Edison Company's plant. His experience had not been very satisfactory in that direction, and he wanted to know if the firemen had any other apparatus to fall back on in case the flow meters failed to give the proper records or not.

I. E. Moultrop said that in times past, managers of plants took an interest in their engine rooms, and let the firemen run the fireroom. But in the fireroom were to be found the greatest possibilities of economy in the operation of the power plant, and it should be in charge of the most intelligent men in the operating department. It was gratifying to see that the Detroit Edison Company was working along that line.

In answer to Mr. Cary, he said that he had had considerable experience with steam flow meters in Boston, and thought that they were quite accurate. As to the matter of checking, there was no instrument in the plant that was good for much if it was not periodically and systematically checked.

He had come to the conclusion that the $CO_2$ recorder was not so valuable an instrument as it had been assumed to be in the past. With feedwater regulators and steam flow meters, and with the intelligent help in the firerooms of today, the $CO_2$ recorder was one of the least valuable instruments on the control board. Mr. Moultrop did not mean, however, to condemn the instrument as a piece of boiler room apparatus. He believed it was a more delicate instrument than others and required more attention to keep it in satisfactory working condition. It had one disadvantage; the other in-

struments mentioned showed what was taking place in the operating of the boiler, whereas the $CO_2$ recorder told what had happened sometime previous, the lag being somewhere between 5 and 20 minutes' duration. The fireman was not especially interested in post mortems; moreover, an intelligent fireman would find ways of getting satisfactory $CO_2$ readings on his recorder without necessarily getting ideal conditions in his furnaces.

George B. Preston had found that one of the principal difficulties in starting up large boilers was in connection with the firebrick lining and the combustible arches, and he wished to ask what material Mr. Parker found to be the best, and if he had any illustrations showing the failure of these brick arches.

Roger DeWolf asked what connection there was, if any, between the blower supplying the air to the stoker and the drive for supplying the coal. He also wished to know if Mr. Parker would consider a semi-automatic control for the blower, at least as being a desirable proposition. A new equipment of 875-h.p. boilers was being installed in Rochester using a semi-automatic control on the blower and a motor-driven feed for the coal, which was entirely in the hands of the firemen for operating.

Mr. DeWolf agreed with Mr. Moultrop regarding the use of $CO_2$ recording meters, but would like to have further details. In the above installation it was being arranged to measure the feedwater continuously in order to get continuous records of the feedwater and of the amount of coal burned, and it was expected to rely on these, together with steam flow meters on the boilers, rather than on any $CO_2$ determinations.

William Kent (written). The $CO_2$ recorder is a necessary evil; it must be used if the best economy is to be maintained. There is no record of tests that I know of (where several tests in succession gave high efficiency) published either in this country or in England, where the $CO_2$ meter, or the Orsat, or the Hempel apparatus for determining the $CO_2$ was not used. I have in mind an apparatus using oxygen instead of carbonic acid gas. The maximum efficiency is coincident with 3 to 7 per cent of oxygen in the chimney gases, and if there is 5 per cent of oxygen in the chimney gases it is as nearly right as it can be. With the $CO_2$ meter on the other hand, if there is 15 per cent of $CO_2$ the economy might be high or it might not, because 15 per cent $CO_2$ means the possibility of a good deal of CO in the gas.

The control of the boilers, as described in the paper, so as to obtain the maximum efficiency, is effected by two things acting together: First, the disappearance of the flame at the top of the combustion chamber, which shows that the combustion of available gas is entirely complete (wherever there is a visible flame there is imperfect combustion; that is, the flame must extinguish itself by burning out so that the combustion is entirely complete to insure that the operation is smokeless). Second, the $CO_2$ should be about 13.5 to 14. The two things are necessarily taken together: If there is smokeless combustion there may be maximum economy, but on the other hand there may be poor economy on account of a possible great excess of air. In order to avoid the error of a great excess of air, therefore, the $CO_2$ apparatus must be used.

Reginald P. Bolton said that these boilers were a wonderful exhibition of what might be expected in the future construction of boilers intelligently designed to give the gases a chance to burn themselves out, giving complete combustion before the gases got out of the boiler. Another lesson to be learned from these boilers was the effect which such large power uses had in improving the morale and the conditions of the working forces.

As to the ineffectiveness of the $CO_2$ recorder, Mr. Bolton was inclined to think that much of the troubles arose from the difficulty of collecting the sample. He had just come from a series of observations conducted on large stacks of the Waterside station of the New York Edison Company, during which a great variety of samples covering a long period of time were gathered from the smokestacks 200 and 300 ft. in height. From a selected point in the stack it was found, after a series of experiments, that the velocities varied all over the stack, that the contents of the stack varied, and in one case, after the medium point had been established by a series of experiments, the current of gases reversed itself in the stack at that point, and there was actually a down-flow instead of an up-flow. It might be well therefore that the poor $CO_2$ recorder as a gage becomes mixed up from time to time. A great deal of study had yet to be made of the flow of gases in flues and smokestacks.

D. S. Jacobus said that careful gas analyses were made in his tests and had not the CO been carefully measured as well as the $CO_2$ it would not have been possible to attain the high efficiencies. The presence or absence of CO was most important. If in operating a furnace it was found that there were certain indications respecting

the length of the flames, or the like, that gave a line on the amount of excess air and the presence or absence of CO in the gases, a furnace could be operated just as well by observing these manifestations as by making gas analyses. Indications of the sort were highly valuable in serving as a guide for securing good combustion. In Mr. Parker's tests both means were employed and contributed to the attainment of the high efficiencies.

Dr. Jacobus complimented Mr. Dow on the progress he was making in putting the right sort of men in the boiler room. He said there was more to be gained or lost in a boiler room than in any other part of a power plant, and he predicted that eventually it would be common practice to place the boiler room in the direct charge of the best men in the organization.

David Moffat Myers said that he had found a very bad state of affairs in the firerooms of factory power plants. The management, as a rule, had not the slightest understanding of what was going on. There was usually not the best of feeling between the chief and firemen, and this was the first thing to be readjusted by the education, not alone of the firemen, but of the management. Mr. Myers had in mind a typical small factory plant which had been drifting along for a good many years in the belief that no improvement could be made. A test showed that two boilers were being fired at only half their capacity, although on certain days they were not holding steam because, as the engineer said, the draft was too weak. They were evaporating 6 lb. of water per 1 lb. of fuel. That plant was greatly improved without spending any money on equipment, by educating the firemen to fire properly. Only one boiler was now being fired and with the damper closed one-third of the time to prevent steam from blowing off. One pound of fuel was evaporating 8.5 to 9 lb. of water instead of 6 lb. as formerly, and the fireman of the plant was delighted. He had been educated by practical example, was getting more money for his services and was intensely interested in his work. The management had their eyes opened and were tremendously interested and satisfied with the results. They were making over 40 per cent more steam for the same consumption of fuel and without a dollar spent for new equipment.

The preventable losses that today existed in factory power plants were tremendous. These losses were chargeable fully as much to lack of education as to improper equipment, and the management was directly responsible for both of these factors.

For some reason it was a fact that owners would go to great expense for producing efficiency in other departments of their business while they allowed their power and heating plants to get along in the most haphazard way. A reform of education in this direction would enable the payment of wages to firemen properly commensurate to their responsibility and would at the same time largely increase the net profits of the owner's business.

Such a reform was gradually taking place, and it was bound to succeed universally for it combined at the same time the principles of science and of humanity.

Harrington Emerson, while praising the detailed information given in the paper, objected to the inclusion of violently varying maintenance or supervising costs as pertinent. There were standards for maintenance costs as well as for power costs, for supervision and for rent, and standard allowances should be used in determining final costs, not accidental variations. In the report criticized, one of the boilers cost in one month for maintenance, labor and material, $1.71, and another $253.02. When such great accidental variations occur the conclusions as to cost of product which include them are vitiated.

Railroads can rightly assume a repair cost of $0.08 a mile for repairs (if this is the average), but no railroad can properly include in the cost of a single trip the destruction of a locomotive by explosion, nor yet omit the item of repairs from the cost of a particular run. In all cost accounting reference to standards is needed.

Albert A. Straub (written). Although Mr. Parker has shown by results obtained in three and a half years of operation, that the operation of so large a steam generating unit is not only feasible, but highly satisfactory, from the most important points of operation and maintenance, it seems, nevertheless, that the choice of such a unit must be governed largely by local operating conditions.

Mr. Parker stated that a scale of $\frac{1}{16}$ in. thickness would form in the hot front tubes, but did not state how long it took to form this scale. I judge it was formed between the periods of turbining tubes, but would desire further information on this phase of the subject. That mud will cause leaky tubes not only with $\frac{1}{16}$ in., or heavier scale, but with scale of egg-shell thickness, and not only at high but at moderate rates of driving, is evident, as shown by Fig. 13. These photographs are of a tube taken from the lower row of tubes of a 375-h.p. Babcock and Wilcox boiler equipped with Roney stokers

and having a small combustion chamber. The damaged portion of the tube was exposed directly to the action of the flame. The feedwater was taken from the Allegheny River and contained from six to ten parts per 100,000 of incrusting solids, being treated with enough barium hydrate to neutralize the sulphate content. The tube was a 4-in. No. 10 gage, lap-welded, charcoal iron tube, so placed that the weld was not exposed to the action of the flame. The photographs of this tube show that the leak occurred directly under the deposit of mud and that the scale was so thin as to have practically no effect. Observations made at other times on tubes which were renewed at this station showed mud to be the cause of overheating. The boilers at this station operate usually at rating and, for short periods, at as high as 75 per cent above rating. The question of the behavior of such a sized unit when using bad water is of vital importance to engineers of the Pittsburgh district. I have

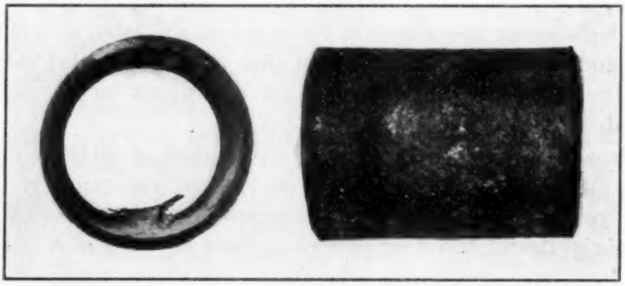


Fig. 13 Cross-Section of Tube, and Outside of Tube Showing Leak

plotted (Fig. 14) 44 analyses of water from the Monongahela River taken at Rankin, Pittsburgh, Pa., from October 14 to November 29, inclusive. Thirty-two of these analyses showed the water to be either neutral or acid as high as 1.50 parts per 100,000. Such a water as this cannot be fed to a boiler untreated and a surface condenser with such a water should not be considered.

With water of this character it has been found that cast iron will be eaten out in anywhere from two to six months, and it is a question what will happen to the tubes of a surface condenser. When using the jet condenser and treating the feedwater in a softener before it enters the boilers a large quantity of sodium salts is fed into the boiler. If the water is treated as it enters the boiler, without a softener, it has been found that a very slight concentration of sodium

salts will cause the water to become wild and hard to keep in the front tubes of this type of boiler, at moderate rates of driving.

Mr. Parker stated that he treated the make-up water with barium hydrate, but did not state where he added it, or whether he filtered

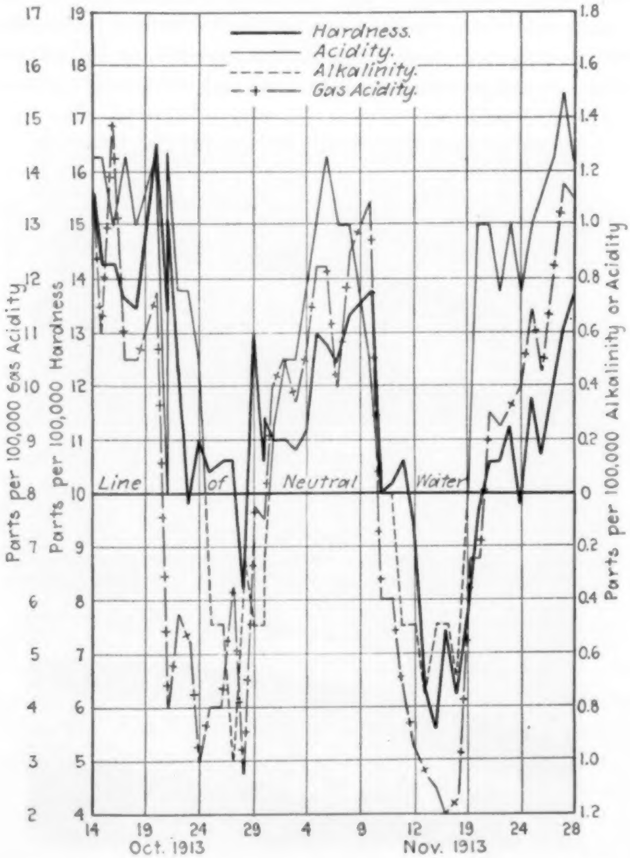



FIG. 14 ANALYSES OF WATER FROM MONONGAHELA RIVER AT RANKIN, PITTSBURGH

the water after it was treated. I have been most successful with this treatment when I could remove the precipitated salts in some manner before the water entered the boiler.

I note no reference to any instrument for recording temperature of escaping gases. This record and that of a recording draft gage

are of vital importance, especially during periods of banked fires, where the overfed stoker is in use.

Considerable stress seems to be placed upon having the proper $CO_2$ content in the gases of combustion, which no doubt is deserving of commendation. My experience has been that it is also advisable to insist upon a low combustible content in the refuse from the ashpit or there is a saving of the one at the expense of the other. Mr. Parker seems to have neglected entirely any reference to the quality

FIG. 15 VIEW OF ARCH IN SAGGED CONDITION FIVE DAYS BEFORE PART OF IT FELL

FIG. 16 VIEW SHOWING ARCH AFTER PART OF IT FELL. FALLEN PART PATCHED

of refuse from the ashpit. I should like to know if he maintains any record of this factor.

In regard to arches on Roney stokers Mr. Parker does not give any data obtained from his experience with sprung arches. Figs. 15, 16 and 17 are views of a sprung arch, which together with a history of this arch, may prove of interest.

The photographs are of an arch of one of a battery of two 375-h.p. Babcock and Wilcox boilers, Alert type of setting with Roney stokers having a great area of 92.5 sq. ft. each. The arch is 11 ft. span, 6 ft. 4 in. long and has a spring of 16 in. The furnace temperatures observed with a Wanner pyrometer through a peep hole 1 ft. below the edge and 1 ft. from the front of the arch, when this furnace was in operation, varied from 2800 deg. to 3000 deg. fahr. This arch was made of Woodland brick and was placed in operation December 12, 1910. On April 18, 1911, the arch was in the condition shown by Fig. 15 (badly sagged), the boiler having been off for leaky tubes

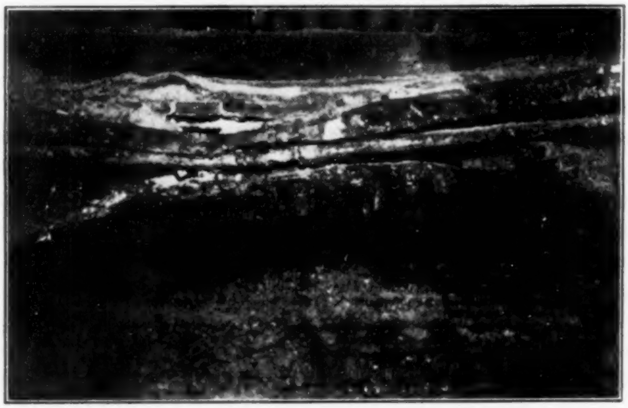


FIG. 17 VIEW SHOWING ARCH AFTER IT FELL COMPLETELY

or general cleaning nine times in this period of four months and six days. The boiler was placed back on the line in this condition and operated until April 23, 1911, when part of the arch fell, as is shown by Fig. 16. This defect was patched and the boiler placed back on the line May 4, and operated without shutdown until June 25, when the whole arch fell, as is shown by Fig. 17. The short life of the arch was due to faulty side walls. The arches and side walls in both boilers were then rebuilt and held from August 1, 1911, to October 29, 1912; during this period the boiler was off 15 times either for leaky tubes or general cleaning.

THE AUTHOR. As far as the fireman is concerned, the steam flow meter need give merely relative indications for his guidance. Extreme accuracy is not necessary for his purpose nor would a con-

stant multiplying error mislead him seriously. The experience at the Delray plant has been, however, that, properly installed and calibrated, the flow meter is reliable and use is being made of it for *absolute* measurements of steam also.

It should be understood that the curve (Fig. 4) of over-all efficiencies obtainable with the underfeed stoker, is a straight line drawn through points taken from the Jacobus tests. The actual efficiencies attained at the extreme points of the range of the tests are given in the paragraph on flexibility in the present paper. The curve of efficiencies presented by Dr. Jacobus in his report of the tests, was intended to show the characteristics of the boiler as fired by either the underfeed or natural draft stokers. The latter tended to give slightly better results at low ratings and lower efficiencies at very high ratings, than did the underfeed stoker.

The efficiency curve of the present paper has been set up as an ideal toward which to work in daily operation. The instruments on the gage board were chosen to give the fireman as much information as possible to aid him in approximating such maximum efficiencies. During a test he has other information continually at hand. Most important is the apparent evaporation, figured cumulatively from the start of the test. However, apparent evaporation is but a derived result and does not show the fireman the causes which lead to it. For judging of present conditions, it is invaluable for him to know drafts at various points of the furnace, rate of steaming as shown by the flow meter and the record of $CO_2$ in the flue gases coupled with observation of the gases entering the superheater. It has been asked why use is not made in this connection of pyrometer readings of the temperature of the exit flue gases. This has been attempted, but for everyday work this exit temperature was found to be dependent upon too many other variables whose influence on it is not well enough known. During a certain test, when the rate of steaming was being held practically constant and the combustion of volatile gases was known by inspection to be very complete, the temperature of the exit gases was found to be extremely valuable in showing up an excess of air. The temperature invariably rose when the gases contained too much air and fell as the excess was decreased. It was possible in this way to predict changes on the $CO_2$ recorder chart, since the recorder readings lagged at least 5 minutes behind the flue gas conditions. In spite of this experience, it was decided to omit exit gas pyrometers from the list of necessary operating instruments.

The almost complete combustion which was obtained during Dr. Jacobus' tests is not too difficult to obtain every day. Flue gas analyses show almost daily as good or better furnace conditions for hours at a time. While driving these boilers at above 200 per cent of rating, and in one case at about 225 per cent of rating, the gas analyses by recorder and Orsat apparatus and inspection of the gases entering the superheater showed better combustion than that secured while running at equally high ratings during the tests. It is only natural that the firemen should become more skilled than they were three years ago. Again, with the self-cleaning stoker installed with one of the latest of the boilers, routine sampling of the ash during December of 1913 showed an average of 12.6 per cent of combustible in the ash, there being 10.2 per cent of ash in the air-dried coal.

In the discussion, further questions were asked in regard to automatic or semi-automatic control of the air supply to the stoker. Before adopting such automatic control, there is this point to be considered. The load of the plant is varying up and down 1 or 2 per cent continually, the greater variations being as a rule more gradual or else periodic. This is, of course, not considering such sudden peaks as those occasioned by thunderstorms. This sawing up and down of the load will occasion slight variations of steam pressure and this allows our taking advantage of the capacity of the boilers for heat storage in the water. It is not necessary that the rate of fuel combustion be so often changed. It would be a mistake to employ an air supply controller which would attempt to hold constant steam pressure, if in so doing it were continually to disturb the rate of combustion of the fuel.

In regard to Par. 27, it was intended to show the cost of material and labor required for repairing these two units during a period immediately following the season of peak loads, when time was afforded for repairs which might have been difficult to accomplish when load conditions were more severe. Since the plant contains various types of boiler and stoker equipment, it was not desired to use for the above purpose, the average boiler, stoker and furnace maintenance costs for the whole plant, which could have been obtained from the plant operating accounts.

No. 1401

# TASK SETTING FOR FIREMEN AND MAINTAINING HIGH EFFICIENCY IN BOILER PLANTS

By Walter N. Polakov, Philadelphia, Pa.

Member of the Society

The object of this paper is to outline a method for the accurate determination and permanent attainment of maximum economy in daily boiler room practice in the power plant by the task-setting method. The problem is two-fold: First, the determination of the conditions which will result in a maximum operating economy; and second, the determination of the factors which will secure the permanent attainment of these best conditions. The writer recently had an opportunity to reorganize the power plant of a large public utility company, and the methods employed to determine the tasks for the firemen and to attain the improvement in results which they represent in everyday running, met with such success that a report on the subject seems to be warranted.

2 The engineering function of a boiler plant is deliberately to convert the thermo-chemical energy latent in fuel into the volume energy latent in steam, through a process of trans-power, while the commercial function is to generate steam of required quality and in a required quantity at the lowest cost compatible with the circumstances. The problem before the management, however, is to determine and provide the necessary and sufficient conditions, the fulfillment of which will bring about and assure the permanency of the desired predetermined results.

3 The degree of engineering perfection in a boiler plant is generally reckoned in terms of thermodynamic efficiency, but this criterion alone is both inaccurate and insufficient. It is inaccurate inasmuch as the inherent thermodynamic efficiency of the plant, if determined either by calculation or by experiment under ideal con-

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

ditions, is not attained in practice; if determined from everyday performance it is short by some unknown quantity, depending upon the variable "human element." It is insufficient since the conception of thermodynamic efficiency of a boiler plant does not include the "time element" which is the chief determing factor in our industrial, commercial and social lives. Thus, the thermodynamic efficiency can represent correctly the degree of perfection of the whole plant only if supplemented by correction factors taking into account the human and time elements, and is only one of many factors to be considered in the problem of reckoning the industrial efficiency of a plant.

4 As in differential calculus, the place of maxima depends upon the limiting conditions of variables and constants, so in our problem the proper and complete solution can be found only by strict observance of the rules of scientific investigation, and the steps to be taken in this research work can be grouped chiefly as follows: (*a*) analysis of elements, (*b*) study of limiting effects of variable and constant elements on the maximum efficiency of partial processes, and (*c*) determination of maximum effect and the correlation of determining factors. When in this manner we learn what can be accomplished and what are the conditions *necessary* for it, we have further to investigate whether these conditions are *sufficient.* Our knowledge would be insufficient if we did not consider the element of human will to attain the task set, and the stimulating factor to keep permanently at work the power of will to maintain these known favorable conditions.

5 The well-known fact that the high thermal efficiency attained by experts during the boiler tests is seldom maintained in everyday practice is due to gross neglect on the part of the management to: (*a*) record the conditions causing the high efficiency during the test, (*b*) instruct the men how to regulate these conditions in order to duplicate the test results, and (*c*) provide an incentive to the men for striving for the purpose desired by the management or owners. To this we may add that in most instances there is no assurance or proof that the high test efficiency was really the highest attainable.

6 The work of testing steam boilers as it is usually conducted is not infrequently done by young college graduates or sometimes by the boiler room force, even in the plants of conservative and reputable companies. The data thus obtained are seldom reliable and complete, and even if every factor, however insignificant, is measured and recorded the results are almost equally worthless, except that

occasionally such tests disclose a serious leak somewhere and enable steps to be taken to remedy the isolated cause of loss. As a rule these evaporative tests are of no further use than to satisfy the owner's curiosity or fancy to see mere figures of efficiency and load and sometimes heat balance. As a consequence the work of testing boilers is brought in this country to disrepute, to the auxiliary role of a "selling trick."

7 A mere heat balance is unable to furnish answers to the many questions of utmost importance for determining of conditions for the most economic steam generation. Lack of positive, well grounded principles as well is responsible repeatedly for faulty construction of furnaces and arrangement of baffles and gas passages. The average data of most tests are of very little value inasmuch as such factors as attention and skill of firemen are dumped into the same heap with the factors of boiler and furnace construction, and of properties of fuel. Furthermore, the most reliable test data are of very limited practical value unless they are so analyzed that the dependence of certain phenomena upon definite factors is clearly revealed.

8 What is needed most for a preliminary analysis is a method of boiler test research which will (*a*) establish a theoretical standard for each partial process and allow the determination of partial and ultimate efficiency, and (*b*) establish the influence of such conditions as variation of specific heat of gases depending upon their composition and temperature, radiation of furnace, variation of efficiency of heat transmission and variation of volume of gases in passages with different conditions of combustion, various rates of firing and percentage of infiltration of air, also influence of superheat on the economy of the process. The graphical method of studying boiler performance as suggested several years ago by Professor Grinevetski, fairly satisfies these requirements, and the diagrams thus obtained, representing the working process of a steam boiler, are the most reliable tools in the hands of an investigator for setting the task for boiler operation independent of the skill of the casual firemen during the test, and plainly establishing the relation between the causes and effects. By the Grinevetski method the relations between the temperatures of the gases and the heat contained in them at various temperatures are shown in the form of parabolic curves, as are similarly the relations between the heat in the gases and the area of heating surface, and also the relations of radiated heat to the heating surface.

9 The development of this method of analysis and its application

to practice, with tests, at the Warrior Ridge power plant of the Penn Central Light & Power Company, formed the basis of the method of work relating to task setting and the maintenance of efficiency, discussed in this paper. These tests were made early in 1913, two of which, known as Nos. 1 and 2, were for the purpose of ascertaining the economic result of the operating method in vogue at the plant

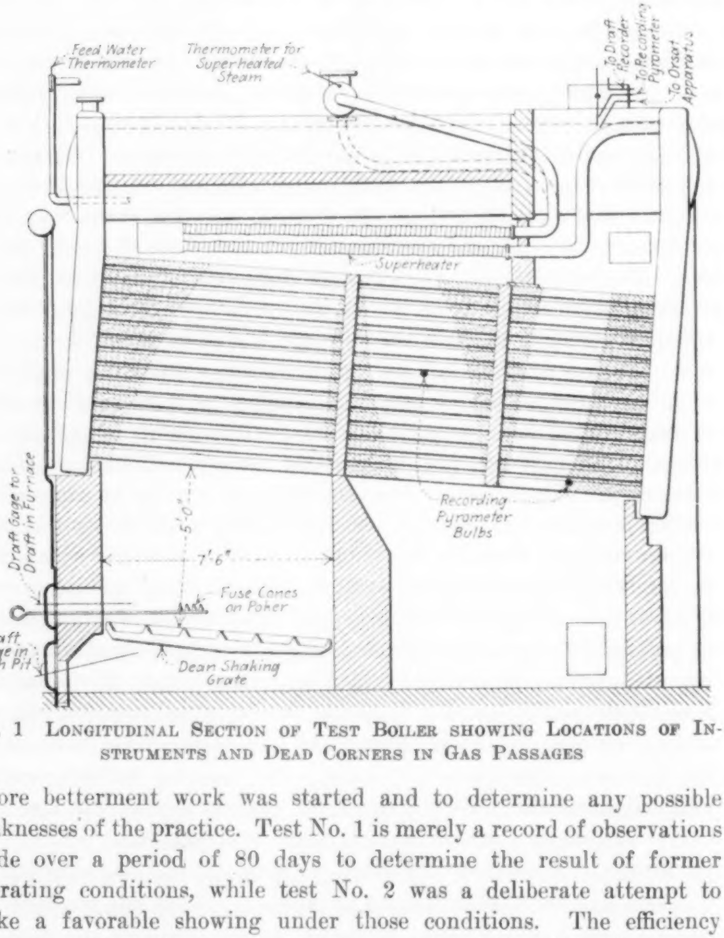



FIG. 1 LONGITUDINAL SECTION OF TEST BOILER SHOWING LOCATIONS OF INSTRUMENTS AND DEAD CORNERS IN GAS PASSAGES

before betterment work was started and to determine any possible weaknesses of the practice. Test No. 1 is merely a record of observations made over a period of 80 days to determine the result of former operating conditions, while test No. 2 was a deliberate attempt to make a favorable showing under those conditions. The efficiency observed in test No. 2 was 67.94 per cent. Other tests, known as Nos. 3, 5 and 6, were run to study the effects of the various changes made in operating conditions and for the standardization of practice, while test No. 6 was a check test on our standard

76.03 per cent when operating at 89.6 per cent of rating, while its official repetition under a load of 115.5 per cent of rating, showed an efficiency of 77.66 per cent. Still other tests were made principally for the selection of coal.

10 The advantages of graphical study of boiler performance were thus plainly demonstrated and the efficiency figure of 77.66 per cent was accepted as a practical maximum of thermal efficiency for this boiler, furnace and grate. As a method, it demonstrates with an appealing clearness the main conditions and results of the process of combustion and steam generation. As a matter of fact the characteristic curve does not change materially in shape for different boilers nor vary at all for the same boiler and same fuel, and thus the problem resolves itself into mere calculation of scales, shifting of coördinates and measurement of projections.

11 The next question therefore was to ascertain what would be the reasonable limit below which no fireman should allow his efficiency to drop. In this we had to consider the factors beyond the control of firemen, viz.: sudden fluctuation of load and unpracticability in that particular instance to analyze gases every 3 minutes; drop of attention every 40 minutes for the period of 11 to 20 minutes and effect of cooling off heating surface and furnace walls during the cleaning of fires (the allotted time for this was set on time studies as 18 minutes per 100 sq. ft. of grate surface). On the other hand there was made a correction for blowing off the boiler once every 24 hours, which credited the boiler with so much metered water and debited it with so many heat units in the same amount of water at the boiler temperature. As a result of these investigations and studies, it was found that the standard for the task should be set at 70 per cent combined boiler and grate efficiency.

12 In this connection it is interesting to note that all officials of the concern and the representative of the boiler maker, considered the task on this basis too high and that accomplishment of it would be out of the question. The only argument that obtained its temporary approval was that it was easier to reduce the task in the future, if not feasible to accomplish, than to make it higher. The fourth month of the task work, however, proved that 73.1 per cent of boiler and grate efficiency was permanently maintained, effecting for the company a gross saving of approximately 25 per cent on the fuel bill alone. This was accomplished, and we would like to emphasize the fact, without heavy capital investment for physical improvement of

equipment like automatic stokers, special grates, force draft fans, soot blowers, etc. The only expense was for a few instruments for the guidance of the firemen in living up to their instruction card.

13 After the test results were carefully studied and the practical maximum efficiency established, it was an easy matter, by referring to the test logs, to standardize the conditions of firing which necessarily result in generating steam at a desired degree of efficiency.

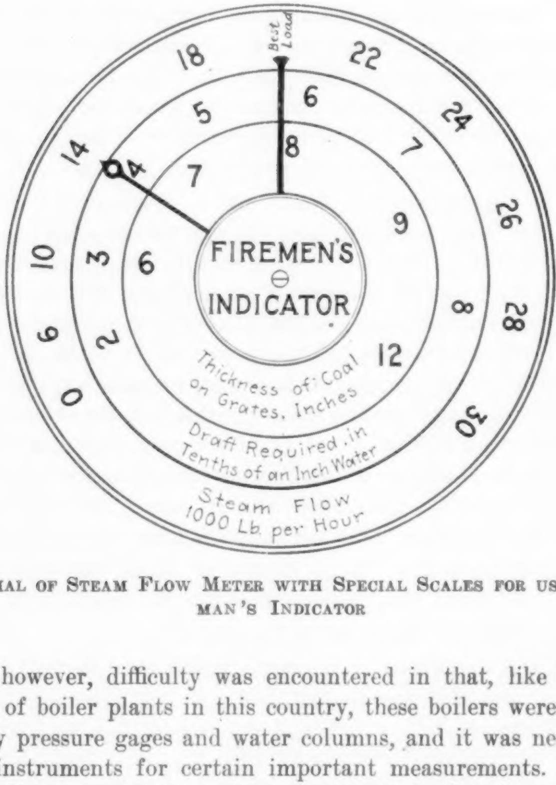



FIG. 2 DIAL OF STEAM FLOW METER WITH SPECIAL SCALES FOR USE AS FIREMAN'S INDICATOR

In this, however, difficulty was encountered in that, like the great majority of boiler plants in this country, these boilers were equipped with only pressure gages and water columns, and it was necessary to provide instruments for certain important measurements. As with other conditions equal, the weight of steam is in direct proportion to the weight of fuel burned, the first instrument necessary for informing the fireman as to how much coal to put in the furnace is a meter indicating the output of the boiler. But as this proportion is constant only at a given condition of combustion, the firemen have to be informed at all times as to what these conditions are. Since complete combustion requires a strictly defined quantity of air per pound of

fuel of given composition, we have to know: (*a*) Composition of fuel; (*b*) rate of firing, lb. per sq. ft. per min.; (*c*) rate of air supply, lb. per sq. ft. per min. In most cases it is possible to obtain fuel of practically uniform analysis, and in addition to have the analysis report within 3 hours from the time of the delivery of coal to the bin, so that the first variable may be taken as a constant. As to the rate of firing, this is indicated by the steam flow meter inasmuch as conditions of combustion are uniform. Then, our last variable, the quantity of air to be supplied per minute to the furnace, is the factor which we have to control.

14 As long as the infiltration of air remains constant (it should be as near to zero as possible) and the cross-section of air ducts or pit doors is constant, the variable elements are the veloctiy of air and its specific weight. The specific weight being a function of temperature, calls probably for not more than two corrections per year for winter and summer average temperatures in the fire room. Thus we have a practical solution of the problem in the indications of a draft gage. Draft, i. e. vacuum over fires, is usually measured in such small fractions of an inch of water column that, considering the resistance of flow of the gases through the boiler as constant for each rate of firing, it is more convenient to take readings at the uptake. From our definition of the draft as vacuum, it follows that, in itself, it is no indication as to quantity of air flowing through the fuel bed and must always be considered in relation to the resistance offered by the layer of burning coal. Careful tests can easily establish the desired interrelation between the quantity of air and thickness of fires, or a duplex draft gage can be advantageously used.

15 This reasoning leads to the conclusion that for practical guidance the fireman needs at least three instruments: (*a*) Indicating steam meter; (*b*) draft gage; (*c*) indicator for the coördination of the condition of firing with the load carried by the boiler at any moment. In his experience the writer found that the well-known General Electric steam flow meter of the indicating type could easily and satisfactorily serve the third mentioned purpose. The writer arranges on the dial of the steam flow meter an inside dial, as shown in Fig. 2, with numbers indicating the required thickness of fuel bed corresponding to the number of pounds of steam drawn from the boiler and a third dial with numbers indicating the draft which is necessary and sufficient to supply the required quantity of air for the combustion at a rate called for by the indicated steam demand. Thus, if the pointer, as in Fig. 2, shows that steam is flowing from the boiler

at a rate of say 14,000 lb. per hour, the fireman will know that the figure 4 under the pointer on the middle scale means that a draft of 0.4 in. of water is needed and that the location of the pointer on the inner scale between the numbers 6 and 7 calls for a thickness of fires of from 6 to 7 in.

16 The next information vitally important for the fireman is the frequency at which his furnace must be coaled to keep the fires

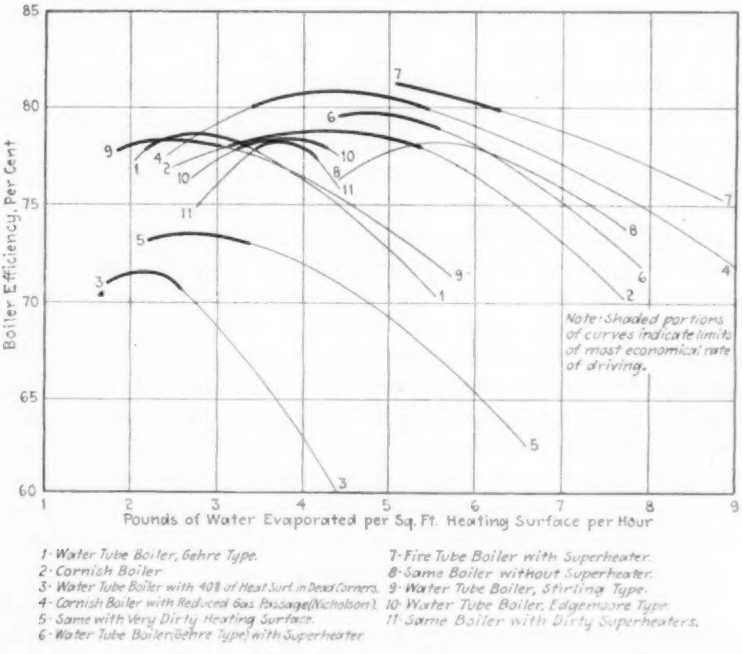



FIG. 3 DIAGRAM SHOWING MOST ECONOMIC LOADINGS FOR VARIOUS TYPES OF BOILERS

in best condition. The method adopted by the Italian navy is accepted by the writer as most satisfactory, consisting chiefly in bell signaling at intervals in proportion to the load carried by the boilers, which signaling is regulated by clock mechanism connected with a flow meter. For use in a boiler house where a number of batteries are fired independently and it was desirable to eliminate the variations of load among them, a modification of this method was devised to equalize the driving of each furnace. For this purpose the counter of the feedwater weigher, supplying water to the entire boiler house,

rings the bell every time a certain number of thousand pounds of water is fed to the boilers, thus giving notice to the firemen that an adequate number of shovelsful must be thrown into each furnace. This number is easily determined since the weight of a shovelful of coal is known and the rate of apparent evaporation at the given condition of firing is a constant.

17 In a public utility plant the load carried by the boilers varies considerably, as a rule, not only throughout a day but often within an hour, in which it may swing from maximum to minimum, or vice versa. This condition was particularly noticeable in the plant of the Penn Central Light & Power Company at Warrior Ridge, Pa., supplying power to the consumers as different in character as residential lighting, coal mines, textile mills, quarries and interurban railroads. While with a number of small steam generating units the peak periods and off-peak valleys in the load curve do not affect the rate of driving individual boilers since the number of boilers put into service can easily be changed, a widely fluctuating load within a short period has its limiting influence on the attainable efficiency of steam generation. The time element in boiler performance, or, in other words, the time required for generation of a certain quantity of steam with the greatest economy, is of utmost importance. Some idea of the relation of factor of thermal efficiency to the rate of driving for boilers of different designs is given in Fig. 3, a chart compiled from data gathered from the experience of the writer.

18 Here again we are confronted with the fact that the thermal efficiency is not a final criterion and it is often the case that high rate of driving, although thermally efficient, is too expensive on account of special costly devices necessary for forcing the boilers or uneven distribution of work on the men requiring extra men idle part of the time. Unwarranted fixed charges in one case and a high pay roll in the other make it financially desirable to limit the output of the boilers. Again, in other cases low rates of driving are abandoned irrespective of the thermodynamic advantages on the strength of financial impossibility for the concern to add new units.

19 The above considerations require for proper solution of the problem a scrupulous research into the interdependence of: (*a*) Variation of fixed charges with various time units required for production of a unit of volume energy in shape of steam; (*b*) variations of thermal efficiency with the variations of the above time unit; (*c*) variations of pay roll, etc.; (*d*) variations of the cost of maintenance, etc.; (*e*) physical limits affecting the quality of steam, as size of superheaters

which at certain point are unable to absorb the heat due to slow convection of heat by dry steam; and (*f*) physical limits of space. Thus, while the increase of output up to certain limit usually increases the commercial efficiency, after this point it begins to fall. And it is in most cases erroneous to assume the maximum thermal efficiency point is a best load to operate the boilers at, due to the complexity of the above mentioned limiting factors.

20 From this short reference to the limiting influence of the variable factors on the final commercial economy we may pass to the problem of final criterion for the determination of the task. The process of steam generation can be conveniently represented in the form of the equation:

$$E_s = E_f \times E_p \times E_m \times E_g \times E_c \times E_t$$

where $E$ indicates the ratio of utilization, or efficiency, and the corresponding indexes signify as follows: *s* steam generation, *f* financial outlay, *p* purchase of fuel, etc., *m* attendance of men, *g* gasification of fuel in furnace, *c* combustion of gases, and *t* transmission of heat of gases to the boiler water and steam.

21 In a given plant $E_f$ is constant; $E_p$ is to be determined previous to the setting of the task for men; the quality, characteristics and cost of fuel are so strictly governed by the construction of the furnace and boiler, by the available draft, and methods of stoking by the coking or baking peculiarities of fuel, that from the few varieties of fuel available on the market, the selection of the most suitable is a mere computation of the cost of fuel for generating say 1000 lb. of steam of given quality. This computation, however, shall follow, not precede the determination of the best condition of combustion with each of the varieties of fuel permanently available on the local market at a given cost including freight rates.

22 Efficiency of gasification of fuel $E_g$ is a variable which in its turn is limited by: (*a*) rate of firing, (*b*) fusing and clinkering quality, (*c*) size of fragments of fuel, (*d*) method of firing, (*e*) frequency of leveling or shaking, and (*f*) thickness of coal bed (provided the design of furnace or grates remains unchanged).

23 Similarly $E_c$, efficiency of combustion of gases liberated from the fuel, is limited by several factors: (*a*) composition and nature of volatile matters, (*b*) proportion of gases to oxygen supplied by air, (*c*) volume of gases developed per unit of time, (*d*) velocity of gas currents, (*e*) volume of furnace, (*f*) distance of the combustion zone

from the heating surface, and ($g$) thickness of fuel bed, or length and shape of torch (for gaseous and liquid fuels).

24 Finally $E_t$, efficiency of transmission of heat, even in the boiler of a given design, varies widely and is positively determined by: ($a$) velocity of hot gases, ($b$) volume per unit of time intact with unit of area, ($c$) temperature of the gases, ($d$) pressure inside of the boiler, ($e$) insulating coating of cool gases, soot and scale deposits, and ($f$) desired quality of steam.

25 A method of computing these partial efficiencies is self-evident:

$$E_g \text{ (efficiency of gasification)} = \frac{\text{heat value in refuse from N lb. coal burned}}{\text{heat value in N lb. of coal}}$$

$$E_c \text{ (efficiency of combustion)} = \frac{\text{heat developed, computed from gas analysis}}{\text{heat in fuel less heat in refuse}}$$

$$E_t \text{ (efficiency of heat transmission)} = \frac{\text{heat in steam generated by 1 lb. fuel}}{\text{heat available for heating surface}}$$

In the last case the temperature of the boiler itself at the given pressure could be taken into consideration as well as the temperature required for the creation of draft. In other terms this equation can be represented as the function of temperatures.

$$E_t = \frac{T - t_s}{T - t_k}$$

where

$T$ = temperature of the gases of combustion
$t_s$ = temperature of gases leaving boiler at uptake
$t_k$ = temperature of boiler water

26 The information these formulae convey is of the greatest importance for influence of judgment as to what partial efficiency and to what extent it could be sacrificed in order to arrive at a desired high standard and meet all other practical requirements and conditions. But, it must not be forgotten that they contain no time element in their composition. To introduce this criterion the boiler output per hour shall be recognized in either of the methods:

$$\text{thermal boiler horsepower} = \frac{\text{heat absorbed by boiler per hour}}{1{,}980{,}000 \text{ ft-lb.}}$$

or

$$\text{mechanical boiler horsepower} = \frac{p \times v}{23{,}760{,}000}$$

where

$p$ = steam pressure absolute per sq. in.
$v$ = steam volume generated in cu. in. per hour

27 This general outline of a method of analyzing the limiting values of variables and determination of conditions necessary for the attainment of certain predetermined results would be incomplete without a reference to the practice of studying the influence of individual variables. The writer is convinced that any boiler trial conducted with the most scrupulous care may leave obscure the influence of a number of the above mentioned limiting factors as well as leave the question uncertain whether the maximum practical efficiency or capacity was really obtained during the test and whether or not the various conditions as observed are necessary and sufficient to duplicate the results at any time. The writer maintains, that as long as the question is not to attain accidentally a mark set at random, but clearly and fully to determine all conditions necessary for maximum economy of operation, for the purpose of setting definite standards, it is imperative to conduct a number of separate observations, though of short duration, dealing in each case with one and only one variable in order to find its limiting effects. Only through an *a posteriori* method of reasoning, through a wide induction from a sufficient number of particular observations, may the probability that is tantamount to certainty be attained.

28 Ascertaining finally the physical elements of the efficiency equation and its value and, eliminating the constants beyond our control, we ultimately obtain a reliable basis for judging the efficiency of the "human element" of the problem which could be expressed

$$E_m = f\ (e)$$

and its limit is reached when the difference between the theoretical efficiency of the process and its actual accomplishment is zero.

29 We now come to the question of how to use this available knowledge based on theoretical research in such a way as to secure the best results practicable in regular service by the task method. In setting a task for firemen, it remains to be determined what the scope of the task shall be. In order to accomplish the purpose it devolves upon the management to accumulate the detailed and exact knowledge of the most favorable conditions to attain results and

make it possible and desirable for every employee to live up to them. It is for the employee, on the other hand, either to create or maintain such conditions as are required in the management's specific instructions.

30 Various schemes have been used as the basis of task setting for firemen which to the writer's knowledge have always created dissatisfaction. Certain of these are as follows:

*a* The cost of steam generated was used for the basis of the task in the boiler room of a large cement plant, and a premium offered for the reduction of this cost, but as firemen have no control over the purchase of fuel, maintenance of equipment, etc., this task involved the standardizing of conditions of combustion, for which no instruments were provided and no definite standard or aim was set before them, and the scheme was soon abandoned.

*b* The high percentage of $CO_2$ in flue gas was adopted as a task basis for firemen in several plants, but the men were not trained nor were they even shown how to obtain it. When they occasionally attained the mark, the question remained undecided whether high percentage of carbon dioxide was coincident with the most economical steam generation or not, and the method proved generally unsatisfactory.

*c* A high percentage of $CO_2$ and low percentage of combustible in the ashes, were factors upon which another attempt was made to specify more definitely the firemen's task. The question remains, however, whether the conditions which the firemen must observe to attain the task and produce gas rich in carbon dioxode and an ash with little carbon, are actually the best for transmitting the heat of the gases to the water and steam and whether at any load the same standard is equally beneficial.

*d* A limit on coal consumption as a task for railroad firemen was favored at one time. This idea, probably the most ridiculous and illogical, soon demonstrated its own weakness and has been almost entirely abandoned.

31 The lack of analytic thought in these instances is remarkable; such important factors as condition of equipment, variable quality of fuel, weather conditions, necessary instruments, complete record keeping, thorough instructions, etc., were disregarded or undervalued. The common cause of failure of such schemes has been the desire

to make a short cut and jump over all preliminary studies, and save the time and trouble of training men in a systematic and thorough manner how to accomplish the task set for them. Should such training be undertaken with bona fide intentions, the instructors themselves would be compelled to discontinue the training as soon as they discovered that the control of the conditions affecting the results is beyond the men.

32 The question of measuring the effect of task accomplishment can be approached from either end, but it is more convenient to figure out the result, and if this is below the standard, turn to the records of conditions and there locate the discrepancy between the required results and those obtained. A gage for measurement of the degree of fulfilment of task conditions set is offered by the factor of thermal efficiency of grate, furnace and boiler which is

$$\frac{\text{heat transferred to steam}}{\text{heat available in fuel}}$$

This measurement involves several corrections for factors beyond firemen's control and neither ratio of apparent evaporation nor boiler efficiency nor efficiency of combustion alone are anywhere near sufficient for the purpose of judging the efficiency of the work of men.

33 Actually to calculate the efficiency of the boiler, furnace and grate is a tedious and comparatively long procedure, and is never made in power plants for a day, shift, fireman or gang of firemen working in team. The writer in his capacity of consulting engineer devised and introduced a comparatively simple method of obtaining a complete record of firemen's performance and to figure their efficiency. This method which has been in vogue for over a year at the the central station at Warrior Ridge, Pa., requires the following record data:

*a* Coal records from store issue tickets and coal passers reports compiled every eight hours

*b* Heat value of fuel determined by bomb calorimeter and value of coal in B.t.u. known for each coal pocket

*c* Amount of water fed to boiler (banked boilers fed separately) ascertained for the same periods

*d* Temperature of feedwater recorded

*e* Steam pressure recorded

*f* Degrees of superheat recorded

34 These data are turned over to the station clerk who proceeds as follows:

*a* From the slide rule shown in Fig. 4, he ascertains the

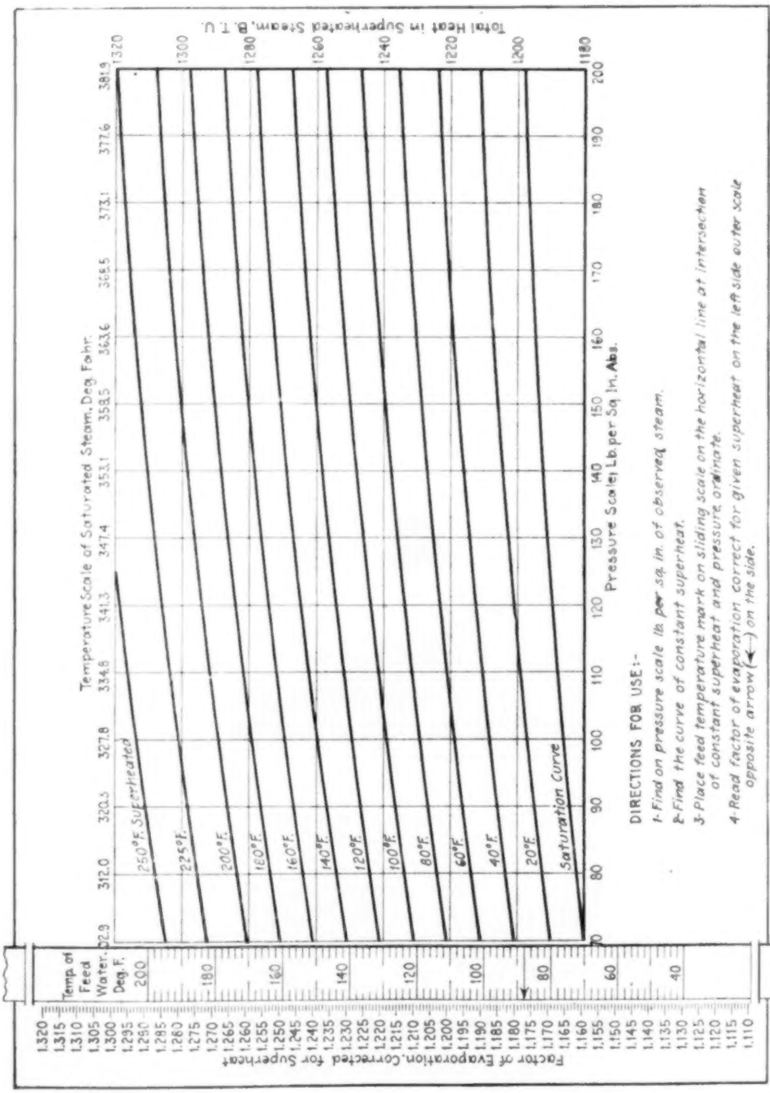



FIG. 4 SLIDE RULE USED FOR DETERMINATION OF FACTORS OF CORRECTION

factor of evaporation (corrected) on the basis of absolute boiler pressure, temperature of feed and temperature of superheat

*b* By means of the power plant log calculator, Fig. 5, he determines

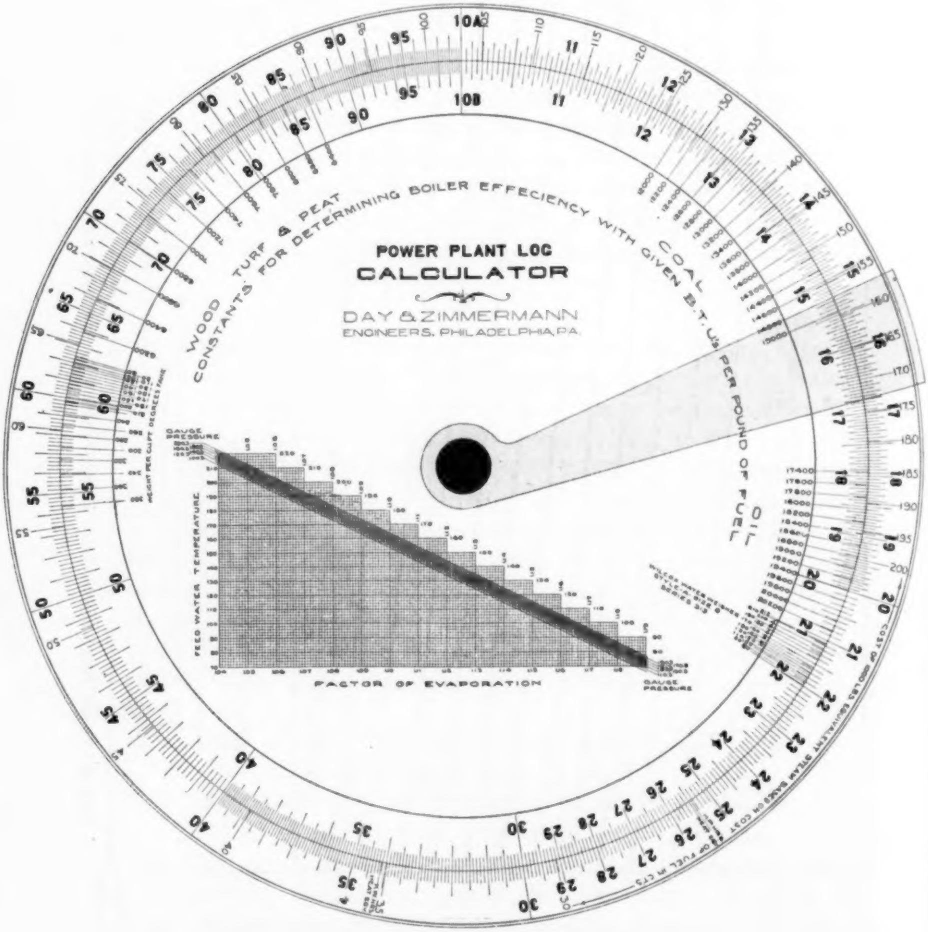



FIG. 5 POWER PLANT LOG CALCULATOR

(1) Actual evaporation ratio during each watch or per each man

(2) Factor of evaporation during each watch or per each man

(3) Equivalent evaporation during each watch or per each man

(4) Efficiency of steam generation per watch or per each man

(5) Cost of fuel per 1000 lb. of steam per watch or per each man

| 9 | 6 | 10/13 |
|---|---|---|
| MONTH | DAY | YEAR |

I HAVE INVESTIGATED AND REPORT BELOW THE
**CAUSE OF LOST BONUS**

DEPT. *Electr.*
DISTR. *War. Ridge*

BY *O. Hawn* ON JOB *Firing*

CONDITION *Draft was set wrong*
*Due to leak in draft gauge*

RESPONSIBILITY IS ALLOCATED TO *Maintenance Man*

REMEDY ATTENDED TO BY ME *Ordering Repair*

*S. D. Port*
SIGNED

| 9 | 12 | 10/13 |
|---|---|---|
| MONTH | DAY | YEAR |

I HAVE INVESTIGATED AND REPORT BELOW THE
**CAUSE OF LOST BONUS**

DEPT. *Electr*
DISTR. *War. Ridge*

BY *D. M. Pope* ON JOB *Firing*

CONDITION *Extra boilers were shut down after midnight.*
*Unexpected load came, too high rate of firing.*

RESPONSIBILITY IS ALLOCATED TO *Switch-board operator*

REMEDY ATTENDED TO BY ME *Reprimanded*

*S. D. Port*
SIGNED

FIG. 6 EXAMPLES OF FORM USED FOR RECORDS OF CAUSE OF LOST BONUS

He then enters the results of computation on the daily power plant report form (see Figs. 11 and 12). The whole procedure takes on the average 18 minutes of the clerk's time, for whom, incidentally, a specific task is assigned and sufficient hourly bonus offered for its fulfilment.

35 Every case of failure on the part of any fireman to secure on his watch the combined boiler, furnace and grate efficiency of 70 per

cent or above is immediately investigated by studies of other records and recording charts of drafts, temperature of escaping gases, nature of boiler refuse, etc., and if no reason can be found there, an examination of the physical condition of equipment and apparatus is made. The result of this investigation is recorded on a form for cause of lost bonus, Fig. 6. This method is particularly valuable and outside of its direct advantages provides an additional and continuous training of the men in careful observation of harmful factors of the slightest nature.

36 Then by means of complete and trustworthy records the firemen are informed as to results of their work before they come back for the next watch (see Fig. 7) and moreover, while they are proceeding

**DAILY BONUS SLIP** CHARGE

DATE *September*

NAME *J. Dearing* NO.

DISTRICT-PLANT *Warrior Ridge*

| | |
|---|---|
| HOURS DAY WORK | *4* |
| HOURS BONUS WORK | *4* |
| HOURS BONUS EARNED | *1* |
| TOTAL HOURS FOR DAY | *9* |

REMARKS *71.8% - 4 hrs. firing on 3rd shift,*
*4 hrs. passing coal for sick coal passer.*

SIGNED *S. D. Port*

FIG. 7 FORM FOR DAILY BONUS STATEMENT TO FIREMEN

with their work, they have in addition to previously mentioned instruments indicating the condition of firing, continuous information as to results they are accomplishing up to any moment of their watch. This is accomplished by having coal weighing and water metering so balanced that an even number of dumps of feedwater and dumps of coal indicates that the ratio of evaporation (superheat, pressure and feed temperature being as specified), is on the safe side of the requirement.

37 The record of attainment of the task by firemen kept in the manner devised by H. L. Gantt (Fig. 8) offers such well-known advantages over any other method that it was adopted for general use

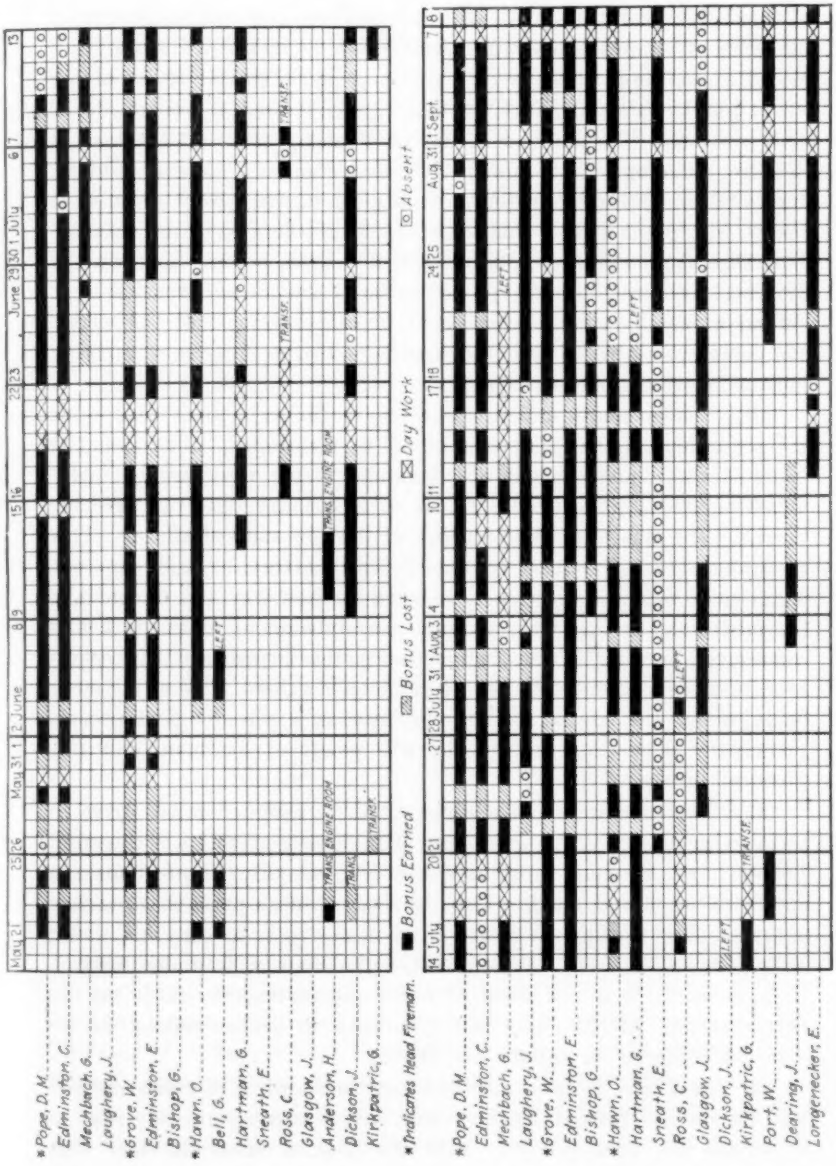



FIG. 8 FORM OF GRAPHICAL BONUS RECORD IN USE

by the above mentioned public utilities companies. The illustration of this record, kept from the start of task work in the boiler house, shows steady improvement and better habits of men. While the May record showed only 68.7 per cent efficiency of boiler and grates of the whole plant, the last month on this record showed the efficiency of 73.1 per cent. The number of day-men falling short on the task is steadily reducing. The number of men absent is, however, chiefly due to "days off" and summer vacations, for the men work seven days a week in a plant of this character. It should here be stated that the departure from the principle of separate man's record made by the writer at first involuntarily proved to be so gratifying, creating as it did an unusually strong team spirit of coöperation, that the writer has never attempted to split the records of two or three men working jointly firing one battery of boilers.

38 The elements affecting the choice for or against task work are so numerous that they cannot all be mentioned. The essential thing is that some element of advantage to the workmen be introduced sufficient to overcome actual or imaginary disadvantages believed by the men to exist as a result of the new state of affairs. This advantage takes the form of a sufficiently attractive and generous bonus to be paid for willingness to learn the new way and to continue to observe the instructions. Actual accomplishment of the task and consequent earning of the bonus means the adjustment of certain conditions concurrent with the adjustment of others. This adjustment being under the control of the employee is a physiological process principally depending on (*a*) the will power of the man, and (*b*) physical fitness.

39 The man to whom a certain task is assigned must strive to accomplish its aim. The exercising of the power of will is a threefold process: first, the man must have a desire; secondly, he must make a choice of ways and means; and thirdly, he must perform necessary actions. The workman's desire from necessity is to earn his living at least; next he has to choose whether he shall work under instructions as set forth in the standing order and instruction cards for the compensation offered; and lastly, he has to act according to his decision in order to satisfy his desire.

40 As a rule, the workmen feel that the adoption of a new method will impose on them an undue strain, but it is comparatively easy to overcome this misconception with the firemen from the fact that greater efficiency means less coal to be shoveled. On the other hand, the new conditions require the men to give their attention to instruc-

tions and the indications of the apparatus, which diverts them unpleasantly from chatting at leisure with their fellow workmen. This forms a more serious obstacle to their quick decision in favor of new routine than anything else.

41 Cases are not infrequent where the men, particularly those more or less in authority, take offense at scientific study and prefer to fight it or even quit rather than admit the advantages of the new practice. Temperament usually determines the vigor of the opposition. Social conditions occasionally influence a man's choice for or against the new regime. If a person who is admired by his associates

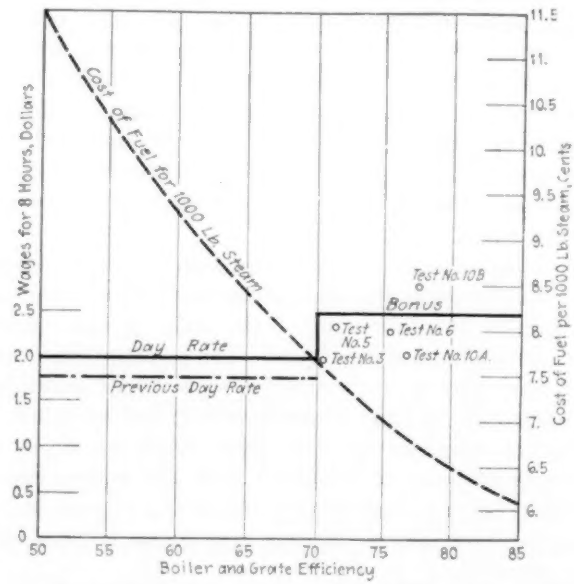



FIG. 9 DIAGRAM SHOWING METHOD OF TASK AND BONUS ADJUSTMENT

happens to decide against task work, he is liable to influence the choice of his friends and acquaintances very materially. Members of labor unions are more apt to disfavor task work, while socialists are strongly in favor of it, feeling that the scientific method of management helps to solve the problem of nationalization of industries.

42 In order to determine the amount of bonus, there must be two limits established—a maximum and a minimum. The maximum bonus should be equal to the amount of net saving accomplished under given circumstances, while the minimum bonus is, of course,

equal to zero. When the bonus to be paid actually reaches either of these limits it loses its usefulness since it loses its stimulating effect—with the management, if the maximum, and with the men, if the minimum. Since in an average boiler house the task results in about 25 per cent saving on the coal bill while the fireman's payroll is from 10 to 15 per cent of the coal bill, it is evident that there is a considerable latitude for adjustment of bonus.

43 After the choice has been made in favor of task work, as a result of the stimulus offered, the third element, proper action, is yet far from being secured. To act in the chosen direction one must know how the desired result is to be accomplished. Lack of sufficient information necessarily produces a strong perception of uncertainty coincident with suffering. After the excellent essay on training workmen by Mr. H. L. Gantt, little if anything can be said on this subject except, that in case of firemen, the success of attainment of the task is determined by detailed, patient, and prolonged training and instructions, and as such this is the most important function of the management. A dummy furnace was found to be an excellent means to break in the green hands.

44 Although the above requirements of additional compensation and exhaustive training for stimulating men's will to coöperate with the management in attainment of the state of high efficiency are imperative, they alone are insufficient. The psychical conditions under which the men have to work must be so arranged as to insure the fullest preservation of their strength, health, and psychical faculties. The opposition exercised by some labor organizations in instances where greater efficiency is demanded from the workingmen without adequate safeguards to their vitality and ability to work is a just and well grounded fight against the short-sightedness of some self-termed efficiency experts.

45 In a boiler house the amount of work per man per hour is constant, and cannot be increased without knocking down the efficiency to a ridiculously low figure, but the number of foot-pounds of work can be reduced in an inverse proportion to the increase of efficiency, so that the question of preservation of a man's health, etc., eliminates any consideration of overspeeding. The conditions which then remain for consideration are (*a*) temperature of room; (*b*) ventilation (dust and draft); (*c*) lighting; (*d*) drinking water; (*e*) restful seats; and (*f*) sanitary washrooms.

46 One familiar with the common layout of a power plant cannot over-emphasize the importance of the above conditions to enable the

men to live up to their task day in and day out. While engine rooms not infrequently offer very pleasant and sanitary surroundings, boiler houses, the most important part of any plant, are so built as to make them unbearably cold in winter and uncomfortably hot during the summer; ventilation apparently serves either to fill the lungs with coal dust or to chill the perspiring men after cleaning their fires. Lighting is an unusual luxury, so that after looking into the furnace no man could read his gages or examine anything around the boiler. Good drinking water is rarely provided, and restful seats with backs

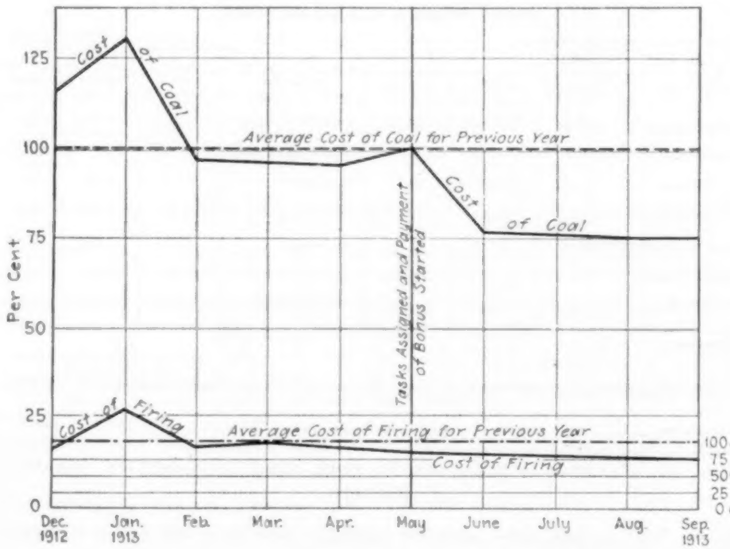



FIG. 10 GRAPHICAL RECORD OF ACTUAL REDUCTION OF COST OF COAL AND LABOR PER KW-HR.

(seats without backs are equal to no seat at all) were never found by the writer in any boiler house. Yet the mere fact that the firemen, if provided with seats having backs, can get through the cleaning of fires in 18 minutes per 100 sq. ft., while without them they consume at best 24 minutes, apparently should convince any unbiased mind.

47 On the contrary, the absence of an elementary condition of comfort in a working place where the men spend the better part of their lives is more harmful to the employers than to the employee. The petty annoyances and feelings of discomfort divert the attention of the men from the performance of their duty to means of avoiding.

the annoyance or harm. Steady attention on the part of the fireman is much more important than is generally realized. A psychological test conducted by the writer on the effect of the occasional drop of attention with the same individual and with individuals possessing different degrees of this faculty proved that *physical condition and strength being constant, the boiler efficiency percentage is in an almost direct proportion to the degree of attentiveness of the fireman.*

48 With healthy, sanitary and pleasant surroundings as a sufficient criterion for judging whether the man's task is harmful, a

DAILY POWER PLANT REPORT

PENN CENTRAL LIGHT AND POWER CO.
STATION Warrior Ridge

FOR 24 HOURS ENDING 9 (MONTH) 14 (DAY) '13 (YEAR) 7.. (HOUR)

| BOILER ROOM | WATCH 1 | WATCH 2 | WATCH 3 | ENGINE ROOM | WATCH A | WATCH B | WATCH C |
|---|---|---|---|---|---|---|---|
| COAL USED (BANKING INCLUDED) | 48800 | 49200 | 34000 | HYDRO ELECTRIC OUTPUT | 980 | 850 | 30 |
| WATER EVAPORATED | 419800 | 408200 | 284000 | STEAM GENERATED OUTPUT | 21220 | 20450 | 15070 |
| ACTUAL EVAPORATION | 8.62 | 8.32 | 8.37 | LOAD FACTOR | 79.5 | 64.4 | 67.5 |
| FACTOR OF EVAPORATION | 1.2187 | 1.2185 | 1.2287 | STEAM PER KWH | 19.78 | 20.00 | 18.85 |
| EQUIVALENT EVAPORATION | 10.50 | 10.13 | 10.28 | COAL PER KWH | 2.30 | 2.40 | 2.26 |
| EFFICIENCY OF GENERATION | 73.4% | 70.8% | 71.8% | THERMAL EFF-CY. OF PLANT | 10.71 | 10.27 | 10.73 |
| COST OF FUEL PER 1000 LBS. OF STEAM | 0.0815 | 0.0845 | 0.0833 | COST OF FUEL PER KWH | 0.00201 | 0.00204 | 0.00200 |

REMARKS:

COAL USED FOR BANKING 1 " BOILERS DURING 6 9 HRS. 55 30 MIN. 3000 LBS. COAL ANALYSIS BTU AVE B.T.U. 13900

TOTAL OUTPUT ____ KW HRS. S. D. Port SIGNED

FIG. 11 EXAMPLE OF DAILY REPORT FROM THE PLANT SHOWING AVERAGE RESULTS

simple and apparently reliable method was devised by the writer. It is evidently beyond the means of ordinary time study to define whether or not the possible output in a unit of time is a safe one from the viewpoint of the preservation of the individual and the nation. The problem in itself is twofold: ($a$) strain per unit of time in foot-pound per square inch of cross-section of a muscle, and ($b$) number of time units at work. As in every case where more than two variables are evolved, the maxima are determined by the limiting conditions of variables, and the constant safe limit can be reached either by a comparatively great strain during a short time or a less intense strain during a longer period. If $t$ and $s$ are time and strain respectively, the equation

$$\frac{d}{d}\left(\frac{c \times s}{t \times c's}\right) = 1$$

if differentiated, illustrates the problem; yet, whether factor $c$ for time or $c'$ for strain has the same numerical value the experiments alone are able to show. More study along this line is needed.

DAILY POWER PLANT REPORT

PENN CENTRAL LIGHT AND POWER CO.
STATION Warrior Ridge

FOR 24 HOURS ENDING 9 | 6 | '13 | 7 -- (MONTH | DAY | YEAR | HOUR)

| BOILER ROOM | WATCH 1 | WATCH 2 | WATCH 3 | ENGINE ROOM | WATCH A | WATCH B | WATCH C |
|---|---|---|---|---|---|---|---|
| COAL USED (BANKING EXCLUDED) | 52000 | 50000 | 31900 | HYDRO ELECTRIC OUTPUT | 1300 | 2240 | 60 |
| WATER EVAPORATED | 449000 | 430000 | 265500 | STEAM GENERATED OUTPUT | 22700 | 22560 | 14840 |
| ACTUAL EVAPORATION | 8.66 | 8.62 | 8.35 | LOAD FACTOR | 88.5 | 68.3 | 62.1 |
| FACTOR OF EVAPORATION | 1.2125 | 1.2159 | 1.2051 | STEAM PER KWH | 19.78 | 19.07 | 18.00 |
| EQUIVALENT EVAPORATION | 10.49 | 10.48 | 10.05 | COAL PER KWH | 2.29 | 2.21 | 2.15 |
| EFFICIENCY OF GENERATION | 73.9 | 73.8 | 70.8 | THERMAL EFF-CY. OF PLANT | 10.85 | 11.22 | 11.54 |
| COST OF FUEL PER 1000 LBS. OF STEAM | 0.0823 | 0.0814 | 0.0848 | COST OF FUEL PER KWH (INCLUDING BANKING) | 0.00198 | 0.00189 | 0.00206 |

REMARKS:

COAL USED FOR BANKING 1 / " BOILERS DURING 6 / 8 HRS 55 / 40 MIN. 5000 LBS. COAL ANALYSIS BTU AVE. B. T. U. 13800

TOTAL OUTPUT ______ KW HRS. S. D. Port SIGNED

DAILY POWER PLANT REPORT

PENN CENTRAL LIGHT AND POWER CO.
STATION Warrior Ridge

FOR 24 HOURS ENDING 9 | 7 | '13 | 7 -- (MONTH | DAY | YEAR | HOUR)

| BOILER ROOM | WATCH 1 | WATCH 2 | WATCH 3 | ENGINE ROOM | WATCH A | WATCH B | WATCH C |
|---|---|---|---|---|---|---|---|
| COAL USED (BANKING EXCLUDED) | 51850 | 54000 | 30750 | HYDRO ELECTRIC OUTPUT | 530 | 4890 | |
| WATER EVAPORATED | 436600 | 423000 | 260500 | STEAM GENERATED OUTPUT | 22470 | 22310 | 13400 |
| ACTUAL EVAPORATION | 8.43 | 7.85 | 8.50 | LOAD FACTOR | 78.3 | 74.4 | 55.8 |
| FACTOR OF EVAPORATION | 1.2125 | 1.2095 | 1.1965 | STEAM PER KWH | 19.43 | 18.99 | 19.46 |
| EQUIVALENT EVAPORATION | 10.22 | 9.49 | 10.17 | COAL PER KWH | 2.31 | 2.42 | 2.80 |
| EFFICIENCY OF GENERATION | 71.8% | 66.7% | 71.5% | THERMAL EFF-CY. OF PLANT | 10.72 | 10.26 | 10.80 |
| COST OF FUEL PER 1000 LBS. OF STEAM | 0.0837 | 0.0900 | 0.0840 | COST OF FUEL PER KWH (INCLUDING BANKING) | 0.00196 | 0.00205 | 0.00208 |

REMARKS:

COAL USED FOR BANKING 1 / " BOILERS DURING 7 / 8 HRS. 05 / 43 MIN. 2000 LBS. COAL ANALYSIS BTU AVE. B. T. U. 13800

TOTAL OUTPUT ______ KW HRS. S. D. Port SIGNED

FIG. 12 DAILY REPORTS SHOWING HIGH AND LOW AVERAGE RESULTS FROM THE PLANT

49 In our experience we adopted in addition to time studies, a careful investigation of fatigue, both mental and physical, and measurements of the vitality of the men affected by various conditions

of work and number of working hours per day. No task is reasonable unless the workman can fully regain his loss between quitting time and recommencing work the next day, and during a sufficiently long period of observation a man should be able to gain or at least not lose anything in his vitality. Observations should cover probably four factors: (*a*) weight of body; (*b*) blood pressure; (*c*) temperature of body; and (*d*) pulse. The blood analysis while considered desirable is too cumbersome to make, and other indications being favorable can safely be omitted. Almost unanimous statements of those observing the task work performed in a great variety of industries assert that men are gaining in health and spirit, but scientifically definite data alone could prove this fact beyond any doubt.

50 Finally, the time element in relation to task setting for men, particularly if the work requires a considerable strain, must be settled by examination no less careful than the study of the time rate of driving boilers. When, however, as in the case of firemen, both physical strain and attention are required, it was found that with strong, healthy individuals the limiting factor on number of hours of profitable work is set not by physical exhaustion but by weariness of spirit. *Other conditions being equal, a fireman on a 12-hour watch is found to be about 4.5 per cent less efficient than the same man on an 8-hour shift.*

51 This time-limiting factor on human efficiency, taken in conjunction with a scientific certainty in determination of the most advantageous thermal efficiency, formed the grounds on which the writer rejected the sliding scale of bonus rate results exceeding the task set by various degrees. The task set must be so little below the most advantageous point that it could be reached with greatest benefit to all concerned and it is not desirable from economical aspects either to fall short of or considerably to overreach it. Offering extra compensation for excess of the task requirement means in final analysis either that the investigator did not determine both limits, or that the management tempts a man to do more than the average employer dares to ask directly.

52 The example of efficient coöperation between employer and employee in the power plants of public utility corporations here referred to demonstrated the value of the above principles for setting task and accomplishing the predetermined results in firing boilers. The diagram in Fig. 10, showing cost per kilowatt-hour of fuel relative to firemen's payroll and bonus before and after adoption of scientific basis for firing, presents, outside of the interesting reduction

in cost since the change of method took place, another feature also of no less importance, namely, that since that time the unit cost remained practically constant, while previously it fluctuated considerably. Samples of daily power plant reports are shown in Figs. 11 and 12.

53 The writer does not claim to have made any new discovery or exhaustive treatment of the technique of the question. The undertaking of presenting to the profession a brief outline of systematic method of task setting for one particular job can be considered fairly fulfilled, if the necessity of thorough scientific research and accumulation of available data for further enlightenment of the many still obscure facts and their coördinations is sufficiently demonstrated.

## DISCUSSION

D. S. Jacobus believed it impossible to get up an equitable bonus system where the men in the boiler room would be paid individual bonuses. The whole organization should be taken together. If an endeavor were made to pay individual bonuses, based on the boiler efficiency secured in each case, the men would soon find out that by carrying a uniform fire they would get the best efficiency. In the case of a fluctuating load those conscientious enough to meet the variations by manipulating the fires to give more or less steam would be penalized through their inability to run at as high a rating as the men carrying uniform fires. It was hard to see how this point could be covered in any equitable system other than by considering the work of the men as a whole.

He considered that the best way of handling the situation would be to put the right sort of a man in charge in the boiler room and make him responsible for the results, in which case the men might be given an increase in pay in proportion to the amount they could save by exerting themselves to the utmost. The general idea that the firemen were opposed to improvements did not seem correct to Dr. Jacobus. He had had considerable experience in working in firerooms to get the best out of the men and knew that if the firemen felt that the expert was in a certain sense one of them, they would do all that they could to coöperate.

H. G. Stott took exception to the curves showing the most economic loadings for various types of boilers. His experience was that the type of boiler had very little to do with that question; it

was entirely a question of the furnace and the combustion of the fuel. A boiler could be driven easily up to 400 per cent rating, or even more. The arrangement of the tubes one way or another was immaterial.

In the table of the rate of boiler readings, No. 7, a fire-tube boiler with superheater, was given as the most economical one. Mr. Stott said he had efficiencies on the regular Babcock and Wilcox boiler equal to those shown in this curve; the Babcock and Wilcox boiler could be carried practically to unlimited rating. Inseparable from this question of rating was the one of the economics of the plant. A plant might be run at an efficiency of 60 per cent during the peak load, and operate more economically, when the fixed charges were taken into consideration, than when running at 80 per cent. When running at 80 per cent, the boilers had to be operated at 70 per cent of the so-called rating, and above that these high efficiencies were not obtainable. At the same time, if two curves were plotted, one of the fixed charges per kilowatt-hour, and the other of the operating cost per kilowatt-hour, it would be found that for the peak load which might last two or four hours a day, the operating charges were relatively unimportant, but the greatest care must be taken to keep down the fixed charges. In other words, the number of boilers on a line must be kept down, as well as the number of banked boilers to be operated.

Mr. Stott believed the time was coming when a boiler plant would be equipped with modern combustion appliances, otherwise modern stokers, which were practically automatic and required little attention. Not firemen, but combustion engineers, or men who had graduated from technical schools, would be required then. It was impossible to get that type of man to go into the boiler room yet, but this would come about when owners of boiler plants were willing to pay more to the man who operated the boiler than to the man who operated the turbine room or the engine room, because the latter could make no change in the economy of the plant, whereas the former could make 5 to 10 per cent difference.

Albert A. Cary (written). The skill of the firemen and the ability or capacity of those who have them under their charge are most important factors in the economical operation of a boiler plant and means must be employed, during preliminary investigation work, to ascertain the part played by such men in arriving at the results obtained. Special apparatus should be used to determine carefully every individual feature included in the furnace, boiler or other part of the equipment and every variable condition which enters into the

operation of the plant should be noted at the time it actually occurs. Merely taking observations during the time of a test and waiting until after its conclusion to work up all results, and then attempting to make deductions, is a practice which will often lead to the overlooking of very important occurrences which would lead to a materially different method of future operation of the plant.

I have found it extremely desirable to use a system in testing by which very frequent gas analyses are obtained (not less than from six to eight per hour) and these results are chalked up on a blackboard as rapidly as they are obtained. By this means it is possible to follow closely the occurrences taking place in the furnace during the course of a test. When making hand-fired tests, I use a device which tells exactly the number of seconds the furnace doors are kept open. Together, these two means serve to size up the value of a fireman very quickly and the comparative merits of two alternate firemen become very apparent in a short period of time. This subject was treated at considerable length by me in a series of papers[1] written for the Iron Age.

I have had considerable experience with Segar cones for determining furnace temperatures and have found that the true furnace temperature could not be obtained through their use on a poker, as was shown by Mr. Polakov. I have found their general use so unsatisfactory that I have abandoned them entirely. The proper use of a good optical pyrometer will give the most reliable results.

Referring to the statement as to the impracticability of getting a gas analysis every three minutes, I have not only obtained the ordinary gas analysis (including $CO_2$; O; CO and N) for 3 minute intervals, but have also succeeded in obtaining the complete analysis of the gas including H and $CH_4$ over 3 minute intervals. But generally, in refined boiler testing, intervals between analyses should not be over 10 minutes.

Mr. Polakov states that the weight of the steam generated is in direct proportion to the weight of fuel burned. I hardly think that he intends this statement to cover operation below rating, at rating and when the boiler is being greatly forced. Also he speaks of having the firemen carry the fire bed at a uniform thickness; this is hardly practical in hand-fired practice where, after cleaning, a comparatively thin bed is started which is increased in thickness up to the time of the next cleaning, although the use of good shaking grate bars will remedy this condition somewhat.

[1]Iron Age, vol. 90, p. 832; 900; 1012; 1026.

Mr. Polakov showed a large number of factors used to arrive at the useful result obtained in steam generation, but these only serve to complicate and obscure a rational consideration of the problem. I make it a practice to determine the individual efficiency of the furnace, and study my furnace conditions apart from the balance of the combination. I then determine the individual efficiency of the boiler and study it by itself, and find whether the boiler is operating under improper furnace or other conditions.

Unfortunately, boiler efficiency is not a fixed quantity under varying conditions of operation, depending largely, as it does, upon the operation of the furnace; with no fire in the furnace, the boiler has no efficiency at all. But, during the course of any particular test, the boiler has its individual efficiency for the set of conditions found at that time, and the study of its efficiency in connection with its controlling conditions shows us where we can improve the defects under which it is operating. These, if remedied, will raise the efficiency of the entire steam generating unit.

The efficiency of the boiler has long been considered and obscured in the reports of *combined* boiler and furnace efficiency, as this combined efficiency is composed of the product of the furnace and boiler efficiencies. When a low combined efficiency results during a test, how are we to know whether the fault is in the boiler or in the furnace? Inference may help us somewhat, aided by the observations taken, but this old time "cut and try" method of making improvements is not in keeping with modern engineering efficiency practice.

R. J. S. Pigott, referring to the diagram showing most economic loadings for various types of boilers, thought there was still a good deal of misapprehension among most engineers as to the economic point of loading, both of a boiler and of a turbine. He found that the efficiency of the boiler and of the turbine had much the same characteristic, namely, there was a dome or high point at some place in the load: with the turbine it was usually at full load that the best efficiency occurred; beyond that point there might be some 25 or 50 per cent capacity allowed for overload. In the boiler this best loading occurred very low in the total capacity of the boiler, about one-third or one-quarter of the maximum capacity. The total rating of the engine room consisted not only of the usual rating of the turbine, but also of the auxiliaries necessary to run the turbine; in other words, the water rate of the turbine might be 12 or 13 lb. per kw. at its best load, but when the load required to drive the auxiliaries was

superimposed, the water rate was much higher, possibly 16 to 18 lb., and the best water rate, including the auxiliaries, was always at a higher load than the best load on the turbine only. This was also true of the boiler room, where there are auxiliaries such as the forced draft fans and possibly induced draft fans, the boiler feed pumps, etc.

While the condenser auxiliaries ran with a practically constant amount of steam, no matter what the load on the turbine, the units in the boiler room were different and required somewhat less steam at lighter loads than at full load. For instance, with a centrifugal boiler feed pump the total steam consumption at zero delivery was seldom less than 55 per cent of the steam at full load on the pump.

If the net output of steam per boiler delivered by the boiler room were considered, the most efficient point would always be at a higher load on the boiler than the best efficiency of the boiler alone.

Edw. A. Uehling (written). Mr. Polakov's paper brings out two very important points: First, it emphasizes the fact that the boiler house is the place where the greatest savings can be made, and second, that boiler efficiency tests are of very little economic value unless adequate means for guidance and control are inaugurated, and due consideration is given to the human element.

The economic conditions found in the plant under consideration in Mr. Polakov's paper are those prevailing in the majority of steam boiler plants. The fact that a "saving of approximately 25 per cent on the fuel bill alone was accomplished without heavy capital investment" for equipment and alteration of plant, etc., except for the necessary apparatus and instruments for control and guidance of the boiler-room crew, should serve to arouse all those power plant managers who have given the question of fuel economy insufficient thought, or have attacked the problem only in a more or less superficial way.

As to the necessity of keeping the fireman informed how well he is, or has been, performing his task, there can be no question; as to the best method of accomplishing this end, opinions may differ. Mr. Polakov presents a very complete bonus system, which is no doubt effective and the principal cause of maintaining the high efficiency reported, but it is too elaborate to merit wide application. I would also question the rationale of his guide to efficient firing controlled entirely by the steam flow meter indicating the thickness of fire and draft to be maintained; there is no constant relation between draft and thickness of fire. The effectiveness of a given draft is modified by several factors: (*a*) the conditions of the fire as affected by the

size of the coal, which may vary greatly from hour to hour, (*b*) the thickness of the ash bed, and (*c*) the presence of clinkers. Thus 0.6 in. of draft may be less effective when the fire is dirty than 0.4 in. after it has been cleaned. Mr. Polakov does not say whether all the boilers are controlled by one flow meter on the main steam line, or whether every boiler is equipped with an individual flow meter. In the former case, how can a fireman tell whether *his* boiler is making its full share of the steam required? In the latter case, the less an individual boiler does, the less the indicator will show what it ought to do. It would seem, therefore, that a guide for the fireman, controlled by a steam flow meter, would be of questionable value. The success which Mr. Polakov scored in attaining and maintaining the high efficiency is due:

- *a* To putting the plant in the best possible condition and selecting the most profitable kind of coal;
- *b* To instructing thoroughly the personnel of the boiler house and providing for legitimate comfort;
- *c* To the inauguration of a liberal bonus system, based on the actual efficiency of every individual fireman;
- *d* To the admirable system of investigating every shortcoming, in order to determine whether it was due to the man, or to something over which he had no control; and if the latter, to the immediate remedying of the cause.

The fact that, in order to carry out the above plan, it is necessary to meter the water correctly to every individual boiler, to keep tabs on the water wasted in blowing off and that fed to boilers banked, and to weigh out the coal for every individual fireman for each shift; also to make the necessary calculations and write out the individual reports, etc., necessarily brings up the question whether equally good results may not be obtained by simpler means.

Mr. Polakov mentioned several other ways in which it has been attempted to inaugurate bonus systems for firemen, all of which he doomed to failure. Among others he mentions the attempt to base such a bonus system on the percentage of $CO_2$ contained in the flue gas. Of this, he says, "The high percentage of $CO_2$ in the flue gas was adopted as a task basis for firemen in several plants, but the men were not trained nor were they shown how to obtain it. When they occasionally attained the mark, the question remained undecided whether the high percentage of carbon dioxide was coincident with the most economical steam generation or not, and the method proved generaly unsatisfactory."

I ask the question: How could it be otherwise? Would not Mr. Polakov's system prove worthless if carried out in the same slovenly way? It can be safely stated that wherever $CO_2$ has failed to be a true guide to efficient firing, it has been due to similar causes.

It is an indisputable fact that of any given boiler fired with the same fuel under the same conditions, the higher the percentage of $CO_2$ within the limits of practically complete combustion, the higher will be the efficiency. Any exceptions that may have been noted are always due to variations in the fuel, either physical or chemical or both, type of boiler, type of furnace, rate of driving, etc.

For example, excessive stack temperature due to hard driving, or insufficient heating surface, or a dirty boiler, may easily neutralize the benefit due to several per cent of $CO_2$, and thus it may, and does frequently happen, that the same boiler driven more moderately, resulting in a low stack temperature, may show a higher efficiency with a lower percentage of $CO_2$ in the products of combustion. When the results from boilers of different types operating under quite different conditions are compared, the apparent contradiction to the above statement may appear even more obvious. None the less the fact remains, that in every case where a comparatively low efficiency is obtained with the high percentage of $CO_2$ it would have been still lower if the $CO_2$ had been lower, and where a comparatively high efficiency results from a lower percentage of $CO_2$ it would be correspondingly higher if the $CO_2$ had been higher, and these apparent contradictions do not all militate against the value of $CO_2$ as a guide for, as well as a control, over the fireman. Wherever exceptionally high boiler efficiency has been achieved and maintained, a high percentage of $CO_2$ in the flue gas has been obtained and maintained.

The high standard set by Dr. Jacobus by the efficiency tests of the large boilers at the Delray plant of the Detroit Edison Company was achieved with a percentage of $CO_2$ averaging over 14 per cent, and this high standard of efficiency has been practically maintained under every day working conditions, using the continuous recording and indicating $CO_2$ meter as the principal guide for the firemen.[1]

That this cannot be otherwise becomes self-evident, when we take into consideration the fact that up to 90 per cent and more of the heat loss in the generation of steam in the average boiler, is due to the heat which passes up the chimney. This loss is the product of weight of the products of combustion, the temperature at which they escape and the specific heat. The weight increases rapidly as the $CO_2$ de-

[1]As reported in the paper by J. W. Parker, Trans.Am.Soc.M.E., vol. 35, p. 307.

creases and the temperature is not materially changed, due to the percentage of $CO_2$ in the flue gas.

Much stress is laid on the danger of losing more by the presence of CO in fuel value due to high $CO_2$, than is gained in reducing the loss of sensible heat by reducing the weight of the flue gas. While I by no means wish to underrate the importance of complete combustion, I do believe that there is much misconception regarding this point. In a paper, Combustion and Boiler Efficiency,[1] read before the Society three years ago, I believe I have demonstrated by some 1100 odd flue gas analyses that there is no relation between the content of CO and $CO_2$ in flue gas; that low $CO_2$ is not a guarantee that combustion is complete and high $CO_2$ is not necessarily accompanied by an appreciable percentage of CO. This fact is also borne out by the flue gas analyses reported by Mr. Polakov, which showed that the lowest percentage of $CO_2$ was accompanied by the highest percentage of combustible gases. Complete combustion depends much less on a large percentage of free oxygen than on the method of firing and the construction of the furnace, and most of all on the furnace temperature, and inasmuch as an unnecessary amount of excess air lowers the furnace temperature, very low $CO_2$ is more likely to be accompanied by combustible gases than rationally high $CO_2$.

It has been abundantly demonstrated that with correct furnace construction, thorough instruction and proper treatment of the firemen, and by providing them with adequate means for their guidance and control, 12 to 15 per cent of $CO_2$ can be regularly attained without appreciable amounts of CO and maximum efficiency thereby maintained. It has also been demonstrated that a large excess of air, i. e., a low percentage of $CO_2$, is no guarantee against incomplete combustion, that sporadic boiler tests and occasional gas analyses are of little value, that half-hearted attempts with an inadequate means for guidance and control cannot succeed and are therefore a waste of time and money.

To insure success the conditions must be studied, radical defects must be remedied, the firemen must be instructed and their legitimate comforts considered, an incentive to excel must be given, and above all, adequate means for their guidance and control must be provided. The correctness of the above general proposition will be conceded by all who have given the question of boiler house economy due consideration.

Dr. Kent made the statement, "If we could get a continuous

[1]Trans. Am. Soc. M. E., vol. 32, p. 1215.

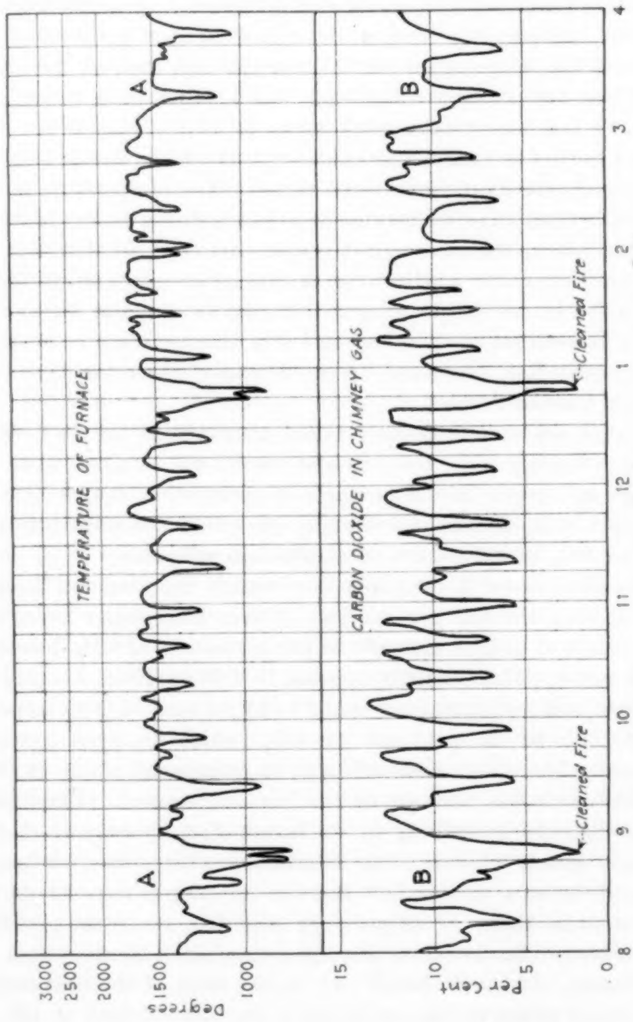



FIG. 13 SIMULTANEOUS RECORDS OF FURNACE TEMPERATURE AND $CO_2$ MADE BY COMBINED $CO_2$ METER AND PYROMETER

record of the furnace temperature, we should have no need of $CO_2$ recorders." This statement is apparently corroborated by the facsimile of two records (Fig. 13), which were made simultaneously by a combined Uehling $CO_2$ meter and pyrometer. The gas analyzed was drawn continuously from a Babcock & Wilcox boiler near the uptake, and the temperature was taken a few feet back of the bridge wall. These two records corroborate one another to a remarkable degree, and one could apparently serve as well as the other as a guide and control of combustion efficiency. It will be noted, however, that $CO_2$ gives the more pronounced record. The temperature is also less valuable than the $CO_2$ because it is not a direct factor in determining the loss up the chimney; it is also unreliable, in that it is impossible to distinguish whether the variations in temperature shown by such a record are due to irregular driving or irregular air supply. But such determinations are impracticable because there is no pyrometer available that can stand up continuously under the high temperature of a boiler furnace.

Dr. Kent has also frequently called attention to the fact, borne out by all efficiency tests, that the best results are obtained when the percentage of oxygen in the products of combustion ranges between 5 and 7 per cent, and he has strongly advocated the substitution of oxygen for $CO_2$ in the control of combustion efficiency.

I have also shown in my paper Combustion and Boiler Efficiency, by coördinating the flue gas analyses of over 1100 boiler tests, that the percentage of oxygen increases as the percentage of $CO_2$ decreases, and vice versa. This relation necessarily follows from Alvogadro's law. When burning bituminous coal 7 to 15 per cent of O corresponds to about 11.5 to 14 per cent of $CO_2$, which is good practice. Theoretically, therefore, it would be quite immaterial which of these two constituents was adopted as the basis of control. Practically, however, $CO_2$ has everything in its favor; first, because it is very easily determined, whereas O is determined with greater difficulty, and second, because of this fact there is no oxygen recorder on the market, whereas there are several very good $CO_2$ recorders available.

There are two fundamental principles on which a bonus system can be established. It can be based (*A*) on the ratio of the heat output in the form of steam to the heat input in the form of fuel, or (*B*) on the ratio of the heat input to heat wasted up the chimney.

(*A*) If based on the boiler output we must know (*a*) the weight of coal fired under every individual boiler and (*b*) its average heat value for each shift and fireman. Second, we must know (*c*) the

weight of water fed into every boiler, (*d*) its average temperature, (*e*) quantity wasted in blowoffs, etc., for every boiler and shift, and also (*f*) the amount of superheat or priming. From these six factors the exact standing of every fireman in terms of efficiency can be calculated and the correct bonus determined.

To determine these data means must be provided (*a*) to weigh out the coal to every boiler for every shift, (*b*) to analyze every car of coal received, (*c*) to meter the water into every boiler, (*d*) to record its temperature, (*e*) to record the temperature and pressure of the steam, and last, the data must be promptly collected, correctly coördinated, and accurately calculated, for which important work a competent clerk must be available for every shift. To be an active stimulus for proficiency every fireman must be informed whether he earned a bonus, not later than the beginning of his next shift.

At the expense of accuracy this method may be materially simplified by assuming (*b*) that the coal is of uniform quality, (*d*) that the temperature of the feedwater does not vary, (*e*) that the quantity of water blown off, etc., is the same for each boiler and shift, and (*f*) that the quality of steam is constant.

(*B*) If a bonus system is to be based on the heat wasted up the chimney, we must know (*a*) the average percentage of $CO_2$ in the products of combustion as they leave the boiler, (*b*) their average temperature, and (*c*) the rate of driving. These three factors give all the information necessary: (*a*) determines the efficiency of combustion, (*b*) shows the efficiency of absorption, and (*c*) tells whether the fireman is doing his full share of the work. These three factors must be autographically and continuously recorded for the purpose of comparing and scrutinizing as well as for the purpose of estimating the bonus. Instruments are now available which will record these three factors on one chart. This reduces the number of charts from three to one, which reduces the time to scrutinize the records, and to average up and estimate their value to one-sixth or less of that required if the data must be taken from three separate charts.

To get the best results it is necessary that the fireman should know what he is accomplishing *all the time*. He must, therefore, have indicators at or near the boiler front which will continuously show the rate at which he is driving his boiler, the percentage of $CO_2$ he is producing and the stack temperature. With this information before him, he can see all the time whether he is operating on a bonus earning scale or not, and this knowledge will not only stimulate, but greatly aid him to get the best results. The pilot steam gage used

at the Delray plant is an excellent idea, where the load varies greatly, as by thus keeping the firemen continuously informed, it obviates the necessity of making a written report to each fireman every day. Estimating the bonus once a week is quite sufficient. This reduces the clerical work required to a minimum.

Determining the bonus according to method (*A*), if fully carried out, has in its favor great exactness; also it shows the daily cost of steam production, and gives the necessary data, by which the coal can be purchased on the heat unit basis, the economic importance of which is not sufficiently realized by coal consumers.

On the other hand method (*B*) has in its favor greater simplicity, more general adaptability, and above all the immediate and continuous occular exposition of the complete process of heat production and absorption, and the continuous autographic record of the three controlling factors, in the best possible form.

The cost of the apparatus and appliances necessary will probably not differ greatly. The cost of upkeep and attendance, including the coal weighers and chemists, will no doubt be largely in favor of method (*B*). Either method if consistently carried out will give maximum efficiency.

The procedure necessary to establish the maximum normal efficiency obtainable in any given plant may vary greatly in detail, but the broad principle outlined by Mr. Polakov must be followed. Whether it is necessary to employ an expert depends on the availability of adequate home talent. Generally it pays to get expert advice. Unless the plant embodies fundamental defects, costly changes are generally not necessary to produce good results. In the majority of cases, it is only necessary to put the plant in proper working order, educate the firemen, or give them the means and the incentive to educate themselves, by inaugurating a bonus system.

Mr. Polakov raised the efficiency of the Warrior Ridge power plant from 54.2 per cent to 78.78 per cent by increasing the percentage of $CO_2$ in the flue gas from an average of 4.3 per cent to 12 per cent. The high efficiency established in this plant can be maintained only as the high percentage of $CO_2$ is maintained; there, as in every other case where high efficiency is attained, $CO_2$ is the most conspicuous single factor.

In conclusion a few words to those who claim that $CO_2$ recorders are unreliable, and they are continually getting out of order. They are neither infallible nor fool proof. In this they do not differ from any other apparatus or machine, a common pump for example. A

steam pump is not a very complicated apparatus and is considered most reliable, but it will not draw water if the strainer is choked up, or a chip gets under the discharge valve. If such a pump be placed in some corner of a plant in charge of *nobody in particular,* it is a moral certainty that this pump would be inoperative the greater part of the time. Nine times out of ten that is the situation where $CO_2$ recorders of modern type (obsolete apparatus are not considered here) fail to perform their legitimate function. In the majority of cases it is even worse because the pump would more than likely stand on neutral ground, whereas the $CO_2$ recorder is generaly projected into a hostile camp. It is looked at askance from the beginning and considered a despicable telltale, at best as one of those new-fangled ideas that won't work anyhow.

In not one case in ten where $CO_2$ recorders fail to perform their proper function does the cause lie in the instrument; nine times out of ten the fault lies with the man higher up, in that he does not hold some one person responsible for its proper and continuous operation. The failure to detail some one and hold him responsible is due either to lax management or to lack of confidence in his own judgment when he decided to install such an instrument.

If the troubles complained of were inherent in the apparatus, it would logically follow that the difficulties must multiply with the number of recorders installed, whereas practically all the failures rumored about, and nearly all the complaints received are from plants where only one recorder has been installed. It is apropos to say here that the benefit derived from the installation of $CO_2$ recorders is in inverse ratio of the completeness of the installation at best. A complete equipment always results in marked savings, whereas a single instrument rarely shows tangible results.

C. A. Austrom (written). In preparing his paper, Mr. Polakov has wisely emphasized the human element as an important factor in the economy of boiler operation. Previous speakers have all agreed with him on this point, but it occurs to me that most of these gentlemen had in mind mainly the larger steam installations, judging from several remarks, as for instance, Mr. Stott's regarding the fact that firing should be in the hands of educated and trained engineers, instead of as in the past by ordinary firemen. In large plants where the firing is done mechanically, this can be accomplished without much difficulty, but in the multitude of small steam plants I believe

that we shall in the future, as well as in the past, see the shovels wielded by the old type firemen and not by college graduates.

Very little has been done to graduate firemen in their line of work, considering the great importance of such an education from the viewpoint of both individual and national economy. The economical end of boiler operation is very little understood by the average fireman and some kind of treatise, dealing with the problems involved in the economical operation of boilers, written for the benefit of these men, has been a long felt need. Such a publication has recently been issued by the Travelers Indemnity Company of Hartford, and this, I believe, is the first attempt made along these lines. It deals with practically all the factors relating to boiler economy and should be of great help to firemen and owners of small steam plants.

Harrington Emerson (written). Efficiency in boiler plants is measured by the standards set. As elsewhere, everything depends on the value of the standards. The rational standard cannot be the ultimate attained in a test run where all conditions were momentarily perfect, as on a race course. Nevertheless the test run of today becomes the expected regular achievement of tomorrow if we are able to eliminate the unstandardized conditions and operations.

Mr. Polakov's paper is exceedingly valuable because it shows ways of standardizing conditions and operations in the boiler plant. The whole paper is so good that I feel less compunction in calling attention to an omission.

Most manufacturing operations consist of the wise combination of materials, work and equipment. In boiler plants manufacturing steam the materials are chiefly the fuel, the equipment is the whole boiler plant, and the attendants are the workers. The performance of the plant depends on the initial quality of the fuel as well as on the way it is used; it depends on the initial excellence of the equipment as well as on the way it is used; it depends on the *initial quality of the attendants* as well as on the way they are used. Before expecting continuous runs of high excellence we would provide for tests of the fuel so as always to have the standard amount of B.t.u. per lb. We would similarly continuously inspect and test furnace settings, boiler joints, clean off scale or deposit inside and out, etc., etc.

How can we expect best results unless the boiler room attendants have initial aptitude and are continuously maintained in effective condition? Mr. Polakov lays stress on fuel of the necessary condition, selection and use of fuel, selection and use of equipment, use of at-

tendants, but omits reference to the selection of men. No one would think of raising a big crop, however excellent the climate, soil and tillage, from inferior seed. Mr. Polakov has emphasized the careful adjustment of work to men, but not the equally important adjustment of men to the work.

Frank B. Gilbreth (written). Four years ago I called the attention of this Society to the necessity of a government bureau for the collective cataloguing and disseminating of knowledge regarding management, that the best methods of each individual of each generation might be recorded for all. Mr. Polakov's paper not only shows a typical advance that is being made in management, but also in the conservation of our natural resources. It shows how workers can be taught to be more efficient, to waste less coal and at the same time how to earn higher wages, and how they can perform their duties without wasting their energy.

What Mr. Polakov says regarding chairs in the boiler room brings up the subject of chairs in general in the industries. Absence of chairs in the past in the workshops has been due to general ignorance on the part of managers, but there is now no longer any such excuse. Every act, every moment of unnecessary standing causes fatigue, and fatigue must be allowed for in the determination of a fair task for each worker.

In our work of installing management I have been greatly impressed with the evidenced unnecessary wasteful hardships due to the absence of recognized rest periods in all industries. Rest periods must be had, and therefore should be recognized, although the manager of one large machine shop said: "Sitting down even when there is nothing for the worker to do doesn't look well, and also makes him lazy."

Many occupations have periods of unavoidable delay, and thus require no specially assigned rest periods; while others have no such unavoidable rest periods, and therefore should have them properly timed and scheduled in order to get the best results, least fatigue, and greatest output.

The Author. In view of the fact that many of the points brought out in the discussion are general in character, I feel justified in emphasizing the main points.

First, the method of management as illustrated in our particular example is based on induction otherwise known as the method of scientific investigation. It is a basis of all modern positive science

and is as old as the Novum Organum of Francis Bacon. Applied to the art of management by Dr. Taylor it is known as "scientific management." The more exact is our knowledge of engineering sciences, psychology, physiology and political economy, the higher is our standard and the higher is the task. The accomplishment of the task is primarily based on the *system of education* of men engaged in the work. For this purpose *each particular case shall be studied individually* and all details—task, methods, implements, bonus, etc., must necessarily be concurrent with the existing conditions and circumstances.

Second, the task and bonus method is a most essential part of the power plant management, being an educational campaign with a reward to those who learn. This makes it unnecessary to hire a college graduate to shovel the coal in the furnace when he can train the others to do it well. Professor Jacobus is in sympathy with those "poor fellows who did not know the method employed by other men" and thereby missed their bonus; he sees the danger which actually lies in the strong incentive for coöperation and improvements. This systematic education of the employees in order to improve their work and remuneration stands in the same relation to the selection of help as the method of using the equipment stands to its purchase.

Yet in the particular case related in my paper the question of selection is infinitely less pressing than that of supply. The location of the plant I have reference to is such that to secure any employee or summon a man to help in case of emergency often amounts to an impossibility, and the problem is not whom to hire but where to find anyone. Under such conditions we must say with all due credit to the importance and desirability of proper selection of employees, that the method of training green, and sometimes unfit men, and turning them rapidly into valuable producers is far more important. The race track is the testing department; no one would think of running a race with a dray horse, but the profit of a trucking business depends upon the skill of the drivers and planning of deliveries, and not on pedigree alone.

Third, in the past too much stress has been laid on the physical perfection of equipment and this one-sided engineering is so well rooted in the minds of average managers that nearly all industrial ills and losses have been believed curable by "patent" devices. Used by untrained and often underpaid and indifferent "hands," they fail to pay for their first cost. It matters not whether after rebuilding the furnaces,

baffles, etc., and changing fuel the various boilers will give the characteristic curves of the same value. The most suitable coal might not be obtainable at the desired cost and the reconstruction might be impracticable under the conditions of service. The question is whether with obtainable fuel and existing equipment we know how to get all we can for the dollar's worth; and if not, to learn how the desired result could be obtained. Unless we teach and stimulate the employee to strive at the high task, the best equipment, if purchased, will again run in a haphazard manner.

And lastly, to utilize the knowledge to the fullest advantage and to control the process at will, the conditions, not results, must be watched and regulated. The instruments will show the changes of conditions, and instruction cards must tell what to do; then if the cause is properly treated, the result is gratifying whether it appears on the $CO_2$ chart, on auditor's balance sheet, or is reckoned in thermodynamic terms.

## No. 1402

# THE PROPERTIES OF STEAM[1]

By R. C. H. Heck, New Brunswick, N. J.

Member of the Society

The fundamental physical properties of steam are pressure $p$, temperature $t$, specific volume $v$, and heat content or total heat $h$, Of these, $p$ and $t$ are taken as independent variables, and equations expressing $v$ and $h$ in terms of them are developed. From these the values of internal energy, entropy, specific heat, etc., can be calculated.

2 Two relations from thermodynamic theory find useful application in connecting volume and heat formulae, namely

$$\left(\frac{dc_p}{dp}\right)_t = -AT\left(\frac{d^2v}{dt^2}\right)_p \qquad [1]$$

and

$$\left(\frac{dh}{dp}\right)_t = -A\left[T\left(\frac{dv}{dt}\right)_p - v\right] \qquad [2]$$

And at saturation, where $p$ and $t$ cease to be independent of each other, there must be satisfaction of Clapeyron's equation

$$\frac{r}{u} = AT\frac{dp}{dt} \qquad [3]$$

where $r$ is latent heat and $u$ is the increase of volume from water to steam during vaporization.

3 The two general equations developed in the paper are as follows:

*a* Volume Equation

$$pv = BT - yp - zp^{2.4} \qquad [4]$$

$B$ is the gas constant for $H_2O$, determined by molecular weight and taken as 0.5956, for units as defined in Par. 4.

[1]This paper is published in abstract only. The complete form is on file and may be referred to in the rooms of the Society.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

*b* Heat Equation

$$y = \frac{[4.66365]}{(t+130)^2} \quad \text{.................. [5a]}$$

$$z = \frac{[11.02244]}{(t+40)^6} \quad \text{.................. [5b]}$$

$$h = h_0 - y'p - z'p^{2.4} \quad \text{................ [6]}$$

$$y' = \frac{[4.40815]}{(t+130)^2} + \frac{[6.75057]}{(t+130)^3} \quad \text{.............. [6a]}$$

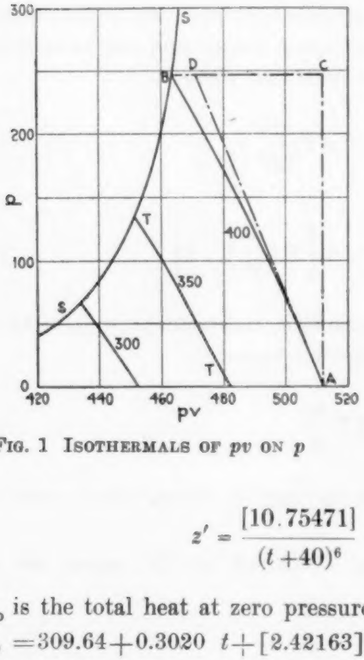



FIG. 1 ISOTHERMALS OF *pv* ON *p*

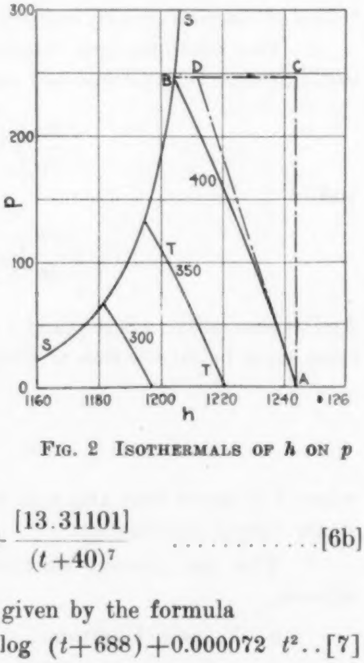



FIG. 2 ISOTHERMALS OF *h* ON *p*

$$z' = \frac{[10.75471]}{(t+40)^6} + \frac{[13.31101]}{(t+40)^7} \quad \text{............ [6b]}$$

$h_0$ is the total heat at zero pressure given by the formula

$$h_0 = 309.64 + 0.3020\ t + [2.42163]\ \log\ (t+688) + 0.000072\ t^2 \text{..[7]}$$

4 These equations are for English units, with $p$ in pounds per square inch absolute, $t$ or $T = (t + 459.64)$ in degrees fahrenheit, $v$ in cubic feet per pound of steam, and heat quantities in the mean British thermal unit. In the paper all formulae are also converted to metric-centigrade units. One special notation adopted is the giving of the logarithm of a constant, thus [4.66365] in equation [5a], in place of the number itself.

5 Equations [4] and [6] are illustrated directly by Figs. 1 to 4,

their derivatives by Figs. 5 and 6. Instead of the volume $v$, which varies so widely with pressure, the product $pv$ is plotted in every case. These three pairs of companion diagrams show the close analogies in behavior between external energy $pv$ and heat content $h$,

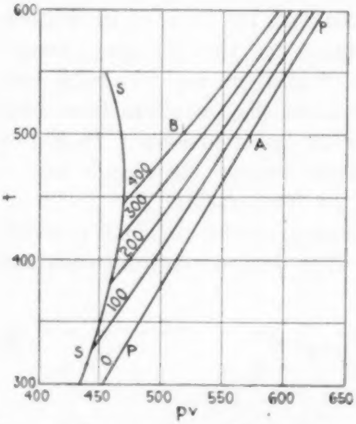



FIG. 3 EXPANSION AT CONSTANT PRESSURE

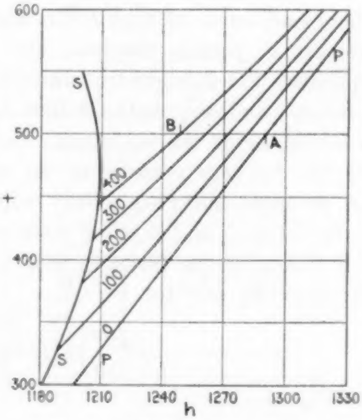



FIG. 4 HEATING AT CONSTANT PRESSURE

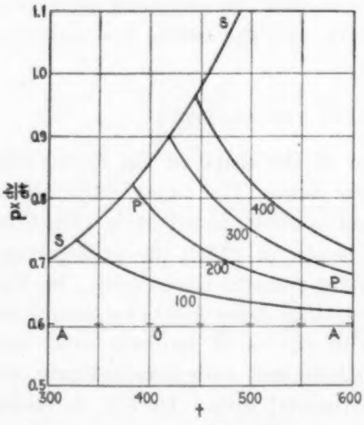



FIG. 5 RATES OF EXPANSION

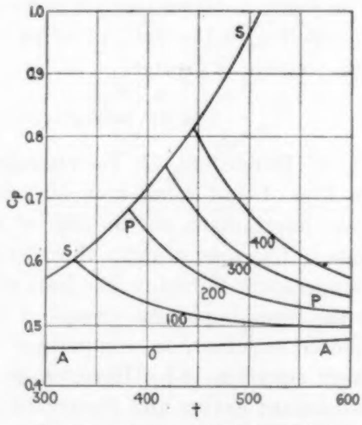



FIG. 6 CURVES OF SPECIFIC HEAT

In Figs. 1 to 6 curve $SS$ is the saturation line, the lower limit of the field covered by the general equations.

6 In Figs. 1 and 2, strong emphasis is laid on the isothermal curves $TT$. For a perfect gas, $pv$ or $h$ would be constant with tem-

perature, or either isothermal would be a vertical line like $AC$. Actually, and especially toward saturation, there is a shrinkage: the straight line $AD$ shows the effect of term $yp$ or $y'p$, while $zp^{2.4}$ or $z'p^{2.4}$ adds curvature to the isothermal.

7 Change under constant pressure is shown in Figs. 3 and 4, where each curve of type $PP$ is marked with the value of its uniform pressure in pounds absolute. In Fig. 3, the line for zero pressure represents the straight-line variation of ideal $pv$ (or $BT$) with temperature; and any distance like $AB$ shows the shrinkage from ideal to actual, along the isothermal and with rise of pressure. In Fig. 4, the line for zero pressure is not straight because the specific heat is not constant but rises slowly with the temperature.

8 Figs. 5 and 6 show rates of change under constant pressure, still retaining the factor $p$ with $v$. The derivatives from equations [5] and [6] are, for Fig. 5

$$p\left(\frac{dv}{dt}\right)_p = B - p\frac{dv}{dt} - p^{2.4}\frac{dz}{dt} \qquad [8]$$

for Fig. 6

$$c_p = c_{po} - p\frac{dy'}{dt} - p^{2.4}\frac{dz'}{dt} \qquad [9]$$

The $y$ and $z$ derivatives are negative so that the secondary terms add themselves to the first, principal term, and the curves are higher as the pressure is greater.

### USE OF THROTTLING DATA AND RELATIONS

9 Heretofore, all determination of the shape of the isothermals in Figs. 1 and 2 has been by volume data. The paper introduces a new determinant on the side of heat content, based upon throttling data. The operation of throttling is one in which the pressure and temperature fall while the heat content remains unchanged. In Fig. 7 the three important groups of throttling experiments on record are plotted together, in comparison with curves of uniform total heat from equation [6]. Here, on $p$ as base and with $t$ as ordinate, are $h$-constant curves and $t$-constant horizontal lines. In Fig. 2, taking vertical $p$ as base and $h$ as ordinate, there are $t$-constant curves and $h$-constant vertical lines. Evidently, the slope of the throttling curves in Fig. 7 must have a direct bearing upon the slope of the isothermal curves in Fig. 2.

10 The relation is established as follows: from the general statement $h = f\ (p, t)$, write the differential equation

$$dh = \left(\frac{dh}{dp}\right)_t dp + \left(\frac{dh}{dt}\right)_p dt \qquad [10]$$

Now impose the condition $h$ = constant or $dh = 0$, and get for the rate of change of $h$ with $p$ when $t$ is constant, or the slope from the vertical of the isothermal in Fig. 2, the value

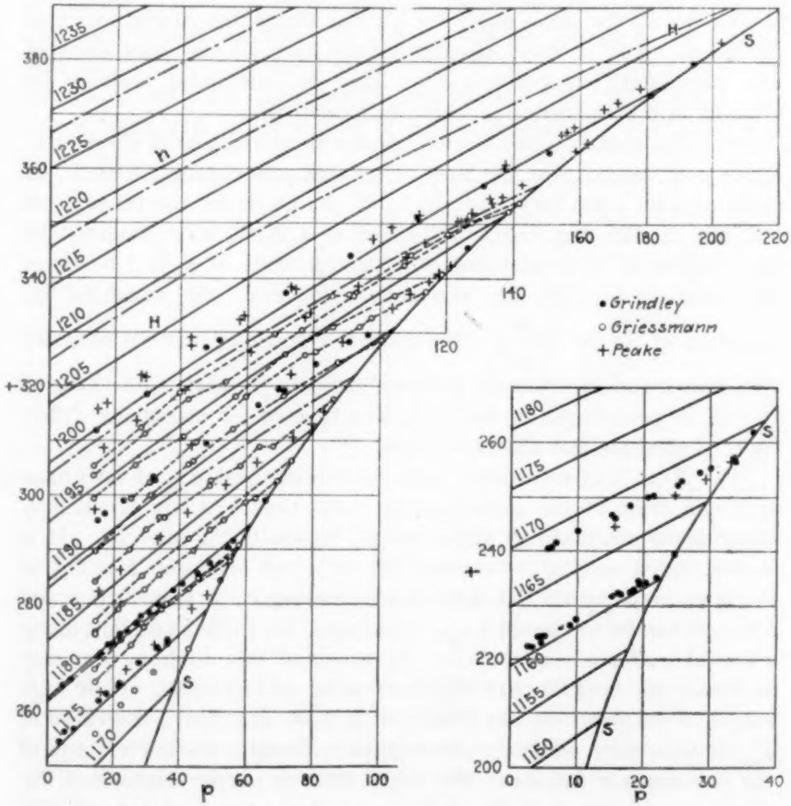



FIG. 7 THROTTLING EXPERIMENTS WITH STEAM INITIALLY SATURATED

$$\left(\frac{dh}{dp}\right)_t = -\left(\frac{dh}{dt}\right)_p \left(\frac{dt}{dp}\right)_h = -c_p\mu \qquad [11]$$

The specific heat $c_p$ under constant pressure is, of course, the rate of change of heat with pressure for $p$ constant, or $\left(\frac{dh}{dt}\right)_p$; and $\mu$ or

$\left(\frac{dt}{dp}\right)_h$ is the throttling coefficient, the slope of the constant-heat curves in Fig. 7.

11 There is considerable experimental information, concerning other gases as well as steam, which seems to indicate that $\mu$ is nearly or quite independent of $p$ and $h$, being a function of $t$ alone. If this is true, any horizontal line of constant temperature in Fig. 7 will be crossed at the same angle by all the throttling curves which it intersects. And if these separate curves have all the same slant at any particular temperature, they may by horizontal shifting be brought into coincidence as one continuous curve.

12 The first step in an investigation along the line of these principles is to assume, for the time being that $\mu$ is constant with $t$. A curve of $\mu$ on $t$ has been laid out by H. N. Davis for the range from 250 deg. to 625 deg. fahr. With this as a start, to be modified by the reaction of trial calculations, a smooth curve of $\mu$ on $t$ has been developed up to 1000 deg. and back to 32 deg. By numerical integration of $\frac{1}{\mu}$ or $\left(\frac{dp}{dt}\right)_h$, the general throttling curve is obtained with base $t$ and an arbitrary pressure ordinate $p'$; that is, the scale of this $p'$, in pounds per square inch, will be used for measuring differences of pressure, not absolute values.

13 Next it is necessary to get the specific heat $c_{po}$ at zero pressure and from it the corresponding total heat $h_o$. The Knoblauch experiments are taken as authoritative, especially the later set. It is shown that a vertical $h$-constant line in a plot like Fig. 2 is cut at the same angle by all the isothermals crossing it, or that with $p$ and $h$ as coördinates the product $c_p\mu$ from equation [11] is constant along a line of uniform heat content. By means of this relation, it is easy to reduce the specific heat observations to zero pressure. Over high ranges of temperature, say from 500 to 1100 deg. fahr., the curve of $c_{po}$ is thus very definitely determined. Toward the lower limit of the experiments (at about 300 deg.) there is poorer consistency, indicating probably that the validity of the method of reduction is weakened by approach to saturation. But there are other data, in Fig. 7 and in the close experimental determination of total heat of saturated steam at low temperatures, which fix very definitely the mean value of $c_{po}$ from 212 deg. to 32 deg. fahr. The curve as finally adopted has the equation

$$c_{po} = 0.3020 + \frac{[2.05941]}{t+688} + 0.000144t \quad \ldots\ldots\ldots\ldots [12]$$

And with the proper constant of integration this leads to equation [7].

14 The method of determining the isothermal or heat content on pressure by means of the general throttling or constant-heat curve is illustrated in Figs. 8 and 9. Curve *HH* is a short portion of the constant-heat curve, $p'$ on $t$. The particular problem taken is that of finding dimensions of the $h$-$p$ isothermal for 500 deg. fahr., up to 300 lb. absolute pressure. In Fig. 8, this isothermal is the vertical line *AB*. From *A* measure down to the scale of $p'$, a series of 50 lb. intervals, and draw horizontal intercepts like *CD*. Then *CD* is the temperature gap (after a drop of 200 lb.) between a $t$-constant line

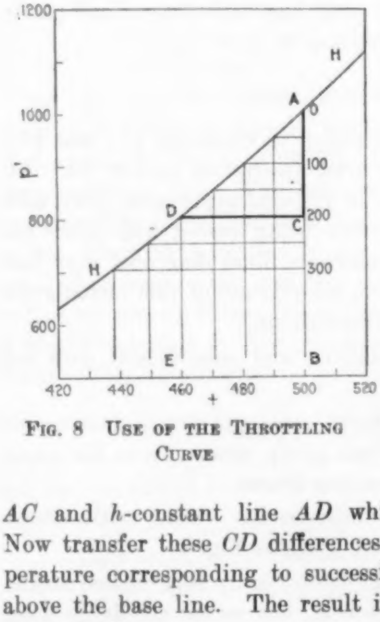



Fig. 8 Use of the Throttling Curve

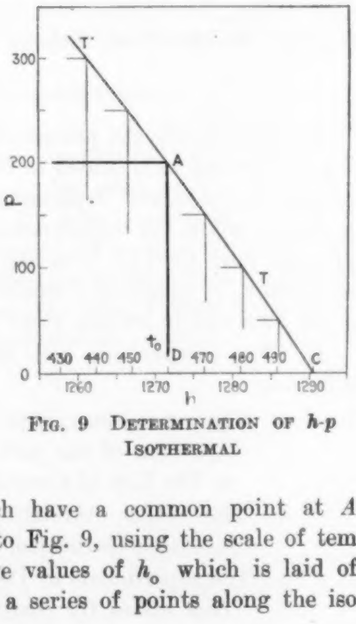



Fig. 9 Determination of $h$-$p$ Isothermal

*AC* and $h$-constant line *AD* which have a common point at *A*. Now transfer these *CD* differences to Fig. 9, using the scale of temperature corresponding to successive values of $h_0$ which is laid off above the base line. The result is a series of points along the isothermal *TT*.

15 This scheme is complete in itself, but the data are by no means accurate enough to make it a complete determinant; nor can the hypothesis that $\mu$ depends upon $t$ alone be accepted as rigorously correct. The method followed has been to obtain by this device a set of isothermals as in Fig. 2, and from them find approximate values of the functions $y'$ and $z'$ and the exponent of $p$ in the last term of equation [6]. Then the task has been to cut and try and adjust constants until equations [5] and [6], tied together by relations [1]

and [2], will satisfy, as nearly as possible, Clapeyron's equation at saturation.

16 One point remains to be considered before discussing the general equations. It is reasonable to assume that the simple law apparently making $\mu$ almost, if not wholly, a dependent upon $t$ alone will have least modification by secondary influences when the pressure is low. From equations [6] and [11] for $p = 0$ is obtained the particular relation

$$\left(\frac{dh}{dp}\right)_t = -y' = c_{po}\mu_o \ldots\ldots\ldots\ldots\ldots\ldots [13]$$

and the curve and tabular column of $\mu$, as given in the paper, is really this $\mu_o$, found after the form of $y'$ had been determined partly by other considerations, through division of $y'$ by $c_{po}$.

## DISCUSSION AND COMPARISONS

17 Concerning the process of developing equations [5] and [6], there is nothing to be added to the brief description in Par. 15. All the data in the several departments of physical experiment were used as checks while the adjustments were being made; and after the formulae were fixed in form and constants, final diagrams were laid out to show, by graphical comparison, the relation of this formulation to the data and to certain other formulations.

18 The comparisons with throttling and specific-heat data are as follows:

- *a* The curve $\mu$ on $t$, with Davis' curve and points showing his collection of the data into group means, from his paper on The Law of Corresponding States
- *b* The classic throttling experiments made with steam initially superheated, here given as Fig. 7
- *c* The experiments of Dodge upon highly superheated steam. This makes for completeness, but serves chiefly to show lack of determinative value, as the experiments show how extremely difficult it is or will be to get reliable values of $\mu$ at high temperatures and pressures
- *d* The Knoblauch-Jacob and Knoblauch-Mollier experiments on specific heat, and the curve of $c_{po}$
- *e* The experiments of Thomas, which show large inconsistencies

In every case, only the formulation of the paper enters into these

comparisons; except that a few lines (dot-and-dash) in Fig. 7 are from Mollier's heat equation.

19 To illustrate the degree of consistency of results by different methods within the investigation, several diagrams are given:

*a* Specific heat $c_{ps}$ at saturation, from equation [9] by proper evaluation and also from Planck's equation

$$c_{ps}=\frac{dh_s}{dt}-\frac{r}{T}+\frac{r}{u}\left(\frac{dv}{dt}\right)_p \qquad [14]$$

The agreement is exceedingly close.

*b* Isothermals from equation [6] in comparison with those obtained by the method of Figs. 8 and 9. There is enough discrepancy to illustrate the fact, as does Fig. 7 also, that complete constancy of $\mu$ with $t$ is not secured in the final formulation.

20 On the side of volume data, the experiments of Knoblauch, Linde, and Klebe, of Ramsay and Young, and of Battelli are plotted on diagrams of the type of Fig. 1, in comparison with isothermals from equation [4]. The first set are the only ones now having determinative value, but they are of the first importance.

21 The real detriment in the close adjustment of equations [5] and [6] to each other was the satisfaction of Clapeyron's relation. In this connection it was found desirable to make a slight adjustment of Marks' equation for the pressure-temperature relation of saturated steam. This formula is

$$\log p=A-\frac{B}{T}-CT+DT^2 \qquad [15]$$

and the two sets of constants are

| | (*a*) Marks | (*b*) Heck |
|---|---|---|
| $A=$ | 10.515345 | 10.606400 |
| $\log B=$ | 3.6878597 | 3.6897500 |
| $\log C=$ | 7.6075880 — 10 | 7.6205462 — 10 |
| $\log D=$ | 4.1439400 — 10 | 4.1601803 — 10 |

The absolute temperature is taken as $T=t+459.64$.

22 These equations are compared in Fig. 10, where straight line *AA* represents equation [15b] and the ordinate is fraction of departure from $p$ as given by this formula. Curve *MM* is the original Marks equation, and the plotted points show the data. A principal reason for making the change is that [15a] makes $p=14.672$ lb. at 212 deg. instead of the proper value 14.697.

23 Various evaluations and tabulations of total heat $h_s$ at satu-

ration are compared in Fig. 11. The base lines *AA* represent equation [6]; but here the ordinate is the actual difference in B.t.u., not the relative difference as in Fig. 10. The curves on the upper *AA* line are concerned with the matter of adjusting equations [5] and [6]. Curve 1 shows $h$ as derived from $h_0$ by the method of Figs. 8 and 9. Curve 2 is the real criterion of consistency between the principal equations, for it represents the total heat derived from volume data, or from equation [5]. Up to 500 deg. or about 600 lb. pressure, the discrepency is everywhere less than 0.5 B.t.u., or barely exceeds 0.03 per cent. Curve 3 shows results of the same calculation, but made with values of $p$ and $r/u$ from the original Marks equation.

24 The curves referred to the lower *AA* line in Fig. 11 are of wider scope. First, curve 4 shows how very nearly $h_s$ from equation [6] is represented by the third-degree equation

$$h_s = 1059.75 + 0.4344\, t + 0.0001829\, t^2 - 0.0000009215\, t^3 \ldots . [16]$$

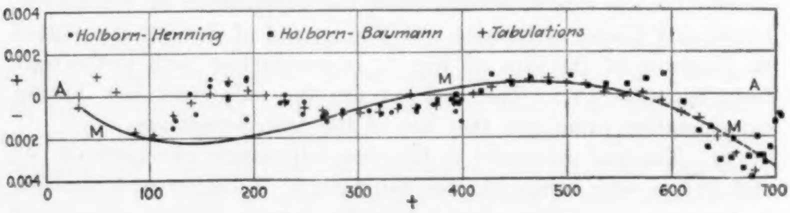



Fig. 10 Pressure-Temperature Comparisons

The curves with numbers higher than 4 represent other formulations, as follows:

No. 5 Marks and Davis *Steam Tables*

No. 6 The writer's table in *The Steam Engine and Turbine*

No. 7 Mollier's *Neue Tabellen und Diagramme für Wasserdampf,* also Smith and Warren's *New Steam Tables*

No. 8 A heat equation from the work of Linde

No. 9 Goodenough's equation, published in Transactions of the Society, Volume 34

25 With these curves are given the experimental results obtained by various investigators: the initials used are, *D* for Dieterici, *H* for Henning, *J* for Joly, *R* for Regnault, *S* for Smith. The Regnault points represent group averages made up by Smith. The points marked *S*1 and *S*2 are of especial interest, as representing A. W. Smith's recent determinations with slow and with rapid vaporization under atmospheric pressure, reported in Physical Review, September 1911. The determining value of $h_s$ at 212 deg. is 1151.2 B.t.u. by

equations [6] and [7] as against 1150.4 by Marks and Davis and 1151.4 used by Mollier.

26 In Fig. 12 a comparison over the practical range of steam pressure and temperature is made between the heat equations of this paper and those of the other formulation which have been put into

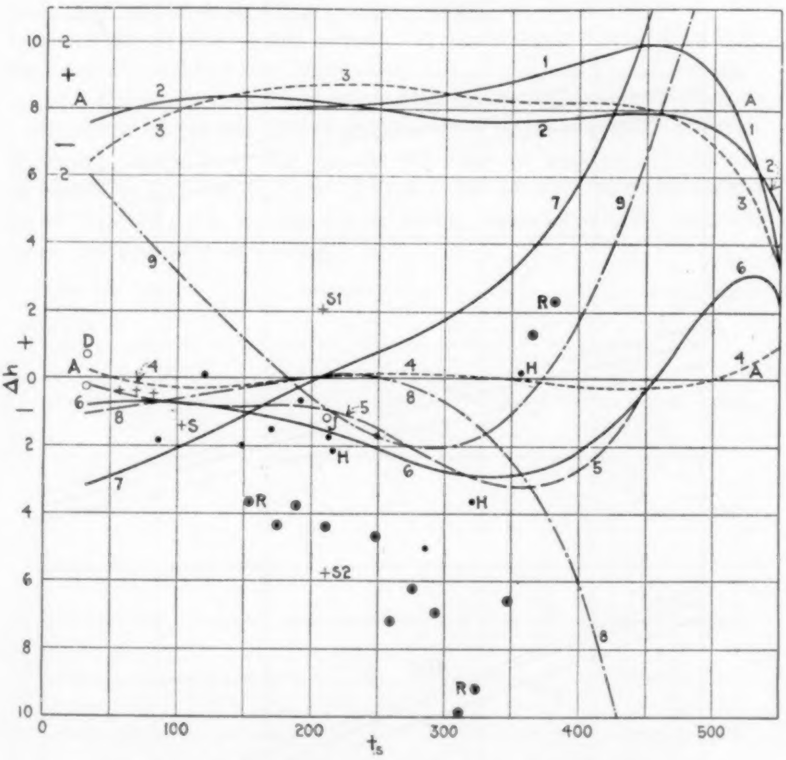



Fig. 11 Comparisons of Total Heat at Saturation

the form of complete steam tables. Full lines 1 represent equation [6], dotted lines 2 the Mollier equation; curves 3 are from the writer's earlier work,[1] points 4 from the Marks and Davis tables for superheated steam. Mollier's saturation curve runs high (compare Fig. 11), and his isothermals are straight lines instead of curves. The Marks and Davis total heats depend upon a wholly empirical and

[1]Steam Engine and Turbine.

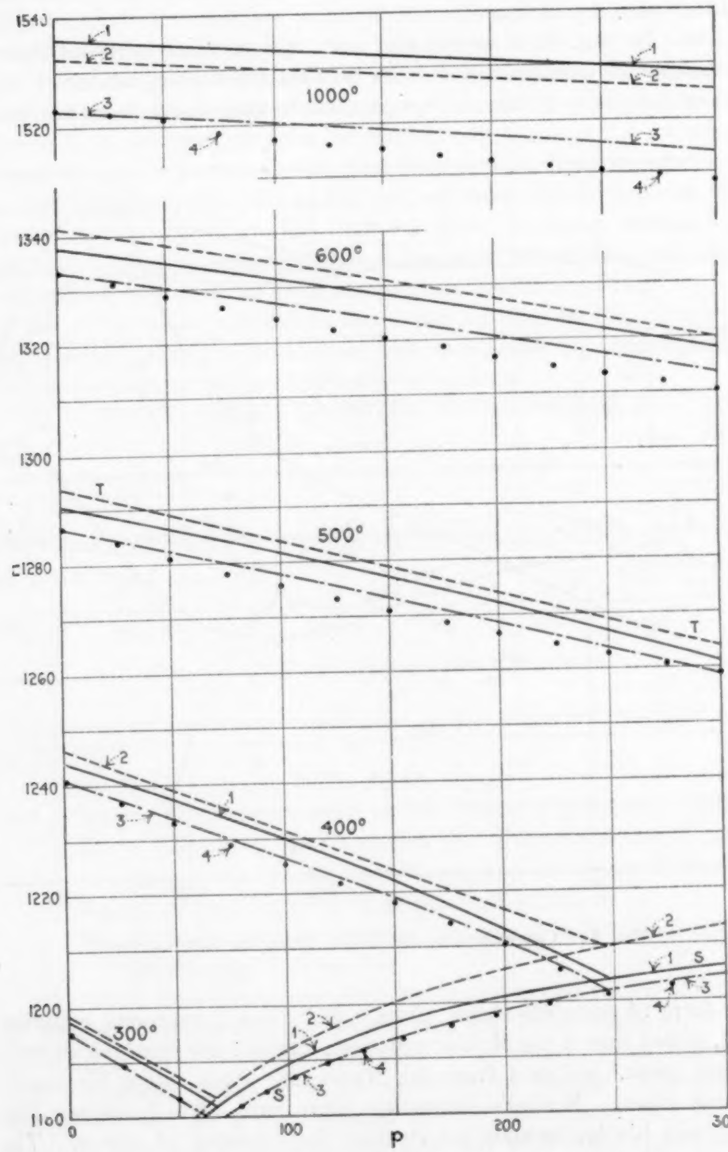



FIG. 12 TOTAL HEAT COMPARISONS

graphical extrapolation of $c_p$ above the comparatively low pressures of the Knoblauch experiments; and the reversed curvature of the isothermals shows that their $c_p$ is relatively too great at high pressures and near to saturation. In the writer's earlier work, a graphical layout and extrapolation of $c_p$ was guided by heat isothermals obtained through the method of the general throttling curve. Dependence upon the Holborn-Henning determination of high range $c_p$ for atmospheric pressure is the reason why isothermals 3 and 4 at 1000 deg. are about 15 B.t.u. below No. 1.

27 Fuller discussion and comparison of the different formulations are given in the paper. To it are appended tables giving at intervals of 10 deg. up to 1000 deg. all the numbers for making calculations by equations [5] and [6] and their derivatives, with pressure factors up to 1000 lb.; also a table for saturated steam at similar intervals up to 550 deg. or past 1000 lb.

## No. 1403

# REPORT ON HOISTING AND CONVEYING

A few years ago, the use of hoisting and conveying machinery was confined almost exclusively to the handling of ore and coal, while today there is hardly an industry that is not interested in the rapid and economical handling of material. High cost of labor and strong competition have increased the importance of this subject and made the material handling engineer an important industrial factor. Improvements in hoisting and conveying machinery have been most rapid where they could be applied without disturbing existing methods, but where radical changes were necessary, progress has been slower.

2 There are now before us far more complex problems that will require in their solution the coöperation of all interested, and your Committee, comprising engineers, manufacturers and users of hoisting and conveying machinery, desires to make itself of real service to the members of the Society as well as to the users of this class of labor saving machinery. To obtain the opinions and encourage discussions along these lines, it begs to submit the following report:

### REVIEW OF THE ART OF HOISTING AND CONVEYING AT THE END OF 1912

3 The hoisting and conveying machine, in which the load after hoisting travels in a straight line, became important in this country in the year 1880. The earliest efficient commercial machines produced after that date were chiefly the invention of Alexander Brown of Cleveland, Ohio. Since that date the improvements have been rapid, particularly in apparatus for handling ore and coal and other bulk materials. The early machines employed buckets that were filled by hand. These have almost entirely disappeared and automatic grab buckets have taken their places. In these machines the winding machinery was installed at one end of the structure where it served to operate the hoisting and conveying ropes.

4 The introduction of electricity as a motive power is directly responsible for the development of the man-trolley machine. In the latter the operator as well as the motors both for hoisting and conveying are frequently carried entirely in the movable trolley.

---

Report of Sub-Committee on Hoisting and Conveying. Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

5 The capacity of the hoisting and conveying machine has been greatly increased, for individual loads handled now reach 50 tons and over, and conveying speeds for lighter loads have increased to more than 2000 ft. per min.

6 The cableway as a hoisting and conveying machine made as rapid progress in its development as those employing rigid tracks supported by bridge structures. The span of cableways, that is the distance between two fixed towers, has reached 2400 ft. Conveying speeds of 3000 ft. per min. have been attained. The self-filling grab bucket and the self-filling scraper bucket are in common use on cableways today. The duplex cableways on power propelled steel towers as used in the construction of the Gatun Locks at Panama represent the highest development of the cableway art. The cableway has found a wide field of usefulness of its own, particularly where portability and semi-portability and long reaches are demarine and railway terminal.

7 Within the past two years the problem of package freight handling by mechanical means has received the attention of railroad and steamship companies, the manufacturers of mechanical handling appliances and public boards interested in terminal construction. Development has been principally along two lines, the overhead and the surface system, though a third, the conveyer system, has received some consideration. The conveyor system, while perhaps well adaptable to certain special conditions, is believed to be insufficiently flexible to meet the complex requirements of a typical marine and railway terminal.

8 The storage battery truck has been introduced and has found favor in several terminals. It takes the place of the old hand truck, making it possible for a man to travel faster and with a much otherwise affecting the problem of hand labor. This truck has the advantage of easy introduction into old terminals requiring only heavier load, hence reducing the cost of transportation but not a reasonably smooth floor. Like the hand truck, it can enter box cars and ships having side hatches.

9 In this connection may be mentioned the inclined elevator which has been introduced at a few marine terminals for carrying loaded trucks up a steep grade. The storage battery truck carrying a light crane has also been introduced, but up to the present time has not found a broad field of usefulness.

10 The overhead monorail system is being pushed by several manufacturers and is looked upon favorably by many interested in

terminal development. It is not a new device but is rather the application of an old device to a new field, for it has long been used to a limited extent in industrial plants. Its introduction into the field of freight handling has resulted in some interesting improvements, especially in track and track switch systems.

11 Ingenious devices have been worked out for throwing the switch to the desired position and automatically placing a stop at the open track end by the approaching trolley. One system lately introduced avoids the use of any moving part in the switch steering mechanism carried by the trolley, enabling the operator to take any desired course through the switch without stopping.

12 An overhead system which has been used to some extent in Europe and which has its advocates in this country, employs what is termed the gliding switch. It has a curved switch member which slides along the main track. It permits a trolley to pass from the the main track to the traveling cross track at any position. This switch is, as far as the main track is concerned, a derailing device. It is believed by the Committee that no installation of this type has been made on this side of the Atlantic.

13 Several installations of monorail systems for handling package freight have been made in this country, one at St. Louis of considerable size and another at one of the railroad terminals at Baltimore. These installations were made previous to 1912 and afford an opportunity to study the problem with a view to determining the type of terminal and layout of monorail system best calculated to produce the desired results of speed and economy.

14 An overhead monorail installation was made at the Baltic terminal, New York, in 1912, which handles freight both ways between vessels, warehouse, and a freight shed alongside a railway track. The system passes over streets and tracks at a suitable elevation so as not to hinder in any way the traffic on the ground level.

15 Another monorail system has been installed in a large wholesale warehouse in Los Angeles, Cal., which combines the handling of package freight from box cars to warehouse and thence for delivering by automobile trucks. A system of elevators permits the trolleys to be transferred to any one of five different floors, and extensions of the tracks over the street make possible the direct loading of automobile delivery trucks. These trucks are designed with removable bodies which are loaded in the warehouse and conveyed by the trolleys to the automobile trucks. Thus a shipment may be loaded on the

fifth floor of a building and carried direct to the delivery automobile truck where it is placed intact without rehandling.

16 The removable bodies are equipped with small wheels so that the trucks themselves may be pushed about like ordinary hand trucks. A system of this type serves as a good demonstration of the saving in freight handling costs that could be made if railroads delivered freight to the address of the consignee.

17 In the present development stage, the storage battery truck and the monorail overhead system appear to be competitors for the field. Certain advantages and limitations of each are recognized. The battery truck as mentioned above will enter box cars and side doors of ships. It will go anywhere a smooth floor is provided, but it will only transport and will not hoist.

18 The overhead system involves greater initial cost. The trolley will go only where track is provided and will not enter box cars, but it will serve flat cars and may reach out over a ship, serving any hatch or the dock without regard to level as affected by stage of tide. It will hoist as well as transport, thus providing a means of transferring freight from one floor to another, and with the traveling cross track will serve areas and tier freight. The monorail trolley affords higher speed of travel than the battery truck. As it does not require clear floor space for a roadway, more floor is available for checking, storage, etc., and the capacity of the terminal materially increased. This increase in capacity should in a great measure, and perhaps fully, justify the greater outlay. It is probably a safe prediction that the two systems, the overhead and the surface, will supplement each other, though different local conditions may sometimes control.

19 Coöperation between railroads, steamship and dock companies, and machinery manufacturers is absolutely essential to the best results in the development of devices for the mechanical handling of freight. Modifications and improvements in the organization and management of terminals must of necessity be made coincident with the installation of the hoisting and conveying appliances in order that such machinery may best serve its purpose.

20 In the past year the mechanical handling of lumber has received considerable attention. This is particularly true on the Pacific Coast, where extensive preparations are being made for handling the large increase in business that it is predicted will result from the opening of the Panama Canal.

21 The unit system of handling lumber is becoming more and

more general. By the unit system is meant the sorting and stacking of lumber in uniform units which may be separated by skids or chocks, and one unit or package handled at a time. Modern hoisting and conveying appliances are equipped with automatic grapples so that these units may be picked up or deposited by the operator in the hoist. The units will range from 3 to 8 tons.

22 In this work the monorail man trolley has been a recent and important development. The locomotive crane has also found a field in this branch of industry and in some cases the locomotive crane of the portal pier type is needed, so that freight cars may pass between the legs of the cranes and be loaded and unloaded by it. This type of crane is designed to travel along the face of the dock for handling lumber directly to and from boats, and by means of a transfer table the crane may be switched to branch tracks for the handling of lumber to and from storage piles.

23 Probably the most radical development in the lumber handling industry is that which the C. A. Smith Lumber Company of California have undertaken. The unit system is employed and the lumber is handled by various types of cranes. Cantilever cranes are used on the dock face for loading specially constructed vessels for the trade, which enable the large 8-ton units of lumber to be stowed intact. This means of direct handling of large units, and the development of special vessels to facilitate handling most nearly approach the history of the development of the coal and ore handling business on the Great Lakes.

24 We find also that during the year 1912 portable cableway log skidders have been produced and many have recently been installed in the mountain regions of North Carolina and Virginia. The features of this apparatus are: (*a*) the power plant is entirely mounted on a special railway car equipped to be coupled into a train of cars and shipped anywhere on a standard gage railway; (*b*) the car carries, besides the operating machinery, boilers, etc., a tower which will clear the tunnels of the railroads. A portable tower is added when the machine arrives on the logging railroad and a steel portable spar, which comprises the head tower of the cableway, is raised when in use and lowered when moving. A tree is used for the tail tower. Spans of 2000 ft. have been successfully employed bringing in logs from inaccessible places and thereby reducing the cost of logging by the saving of railroad building.

25 Two novel forms of cableways have reached their perfected stage during the year 1913. Both of these are for use on the high

seas, one for trans-shipping coal, ammunition and supplies to warships in mid-ocean under headway, and the second for trans-shipping persons from a wreck to a life-saving ship.

26 The apparatus for coaling at sea has been under development for over ten years but not until the last year, and since the newly developed automatic tension engine has been added, can the marine cableway for coaling at sea be regarded as a complete solution of the problem. During recent tests at sea a collier, while rolling 20 deg. in a driving storm, trans-shipped 83 tons of coal in an hour to a battleship 400 ft. astern of the collier while both were proceeding at a speed of about 7 knots.

27 The life-saving machine comprises the old-fashioned breeches buoy apparatus, in common usage along the coast, in connection with a small sized automatic tension engine serving to wind in and pay out the main supporting cable or hawser serving as a trackway for the passage of the breeches buoy from ship to ship.

28 The year 1912 also marked the completion of an extraordinary group of four cableways employed for building the Tunkhannock bridge on the Delaware, Lackawanna & Western Railway. The chief novelty arises in the employment of a center tower 230 ft. high serving as a tail tower for the cableways in line. A further novel feature of these four cableways is in the equalizing apparatus to save the strains upon the guys of the central tower. The hoisting speed is 200 ft. per min. and the conveying 1800 ft. per min.

29 During the year there have been installed on the Sag Channel, a branch of the Chicago drainage canal, three different types of drag line excavators consisting of an elaborate crane with a boom 100 ft. long, capable of digging, hoisting and swinging, and dumping at a point 200 ft. back from the center of the canal. These long arm excavators are reported as showing a capacity of approximately 50,000 cubic yards per month, working 24 hours a day.

30 During 1912, the conveyor field was marked by little change except plants with larger capacities than in the past both in chain and belt conveyors.

31 One development which is of interest was the successful demonstration of the possibility of storing material in open piles on the ground and withdrawing same by tunnel conveyors. The plant in question handled 6-in. limestone which was put into storage by a 40-in. belt conveyor equipped with an automatic tripper and carried by a steel structure. The concrete tunnel under the pile housed a 44-in. belt conveyor and duplex gates in the roof of the tunnel allowed the stone

to flow to the belt by gravity. There were successfully handled through a single duplex gate 1300 tons of stone per hour. This type of storage costs about half of the same capacity in bins and in this case replaced bins on the dock, the stone from open storage being carried to the vessels by 44-in. belt conveyors. Seven thousand tons were loaded without difficulty in six hours. This scheme of carrying a large storage on good ground, thus reducing the cost of the dock, is a distinct step forward, particularly where rapid loading is required and where the supporting of heavy loads involves expensive dock work.

32 The second important development in this industry is the extension of the use of conveyors in preparing copper blast-furnace charge. The old methods of spreading the charge on the ground by barrows or by cars into bins was inaccurate and expensive, while the conveyor method is automatic and a continuous sample being taken, the exact chemical contents of the charge are known. This improvement has reduced the cost of preparing the charge from 10 to 15 cents per ton. Two large installations in Arizona during 1912 involving an expenditure of nearly $400,000 are evidence of the fact that smelter engineers are recognizing the importance of using improved automatic machinery.

33 *Data.* As there are few accurate published data concerning capacities, weights, power formulæ, etc., of conveyors and elevators your Committee would urge the engineers in this field to present such data through the Society. These data would be of inestimable value to the engineers designing plants, enabling them to choose the proper type of conveyor and to pass intelligently upon the designs submitted.

34 These installations do not by any means cover the field of development, and the Committee invites members to forward brief descriptions of other interesting and novel installations in the field of hoisting and conveying.

## RECOMMENDATIONS AND SUGGESTIONS

35 *Boilers.* It is not the desire of the Committee to decry the formation of stringent boiler laws, but it urges that if state and city boiler laws were standardized it would reduce the cost and permit the shipment of a boiler from state to state. It would further make it possible to carry boilers in stock ready for quick shipment. This subject was considered by the American Boiler Manufacturers' Association at its New Orleans meeting, March, 1912.

36 At present we have a national uniform law pertaining to

marine boilers, their construction, inspection, etc. All boilers used in navigation and on navigable waters of the United States come under the Steamboat Inspection Service of the United States Government, with headquarters in Washington. Facts have proved that this national uniform law has been good, and under it there have been few accidents.

37 Locomotive boilers by force of circumstances come under the jurisdiction of the United States Government through the Interstate Commerce Commission. This commission is divided into districts, and the inspection is uniform.

38 The state of Massachusetts has an excellent boiler law, and Ohio has just adopted a similar one. Chicago, Philadelphia and Detroit each have their own ordinances regulating boiler construction, all with the result that boilers built to the requirements of Chicago might not pass inspection in other cities.

39 Your Committee offers the suggestion that if the influence of The American Society of Mechanical Engineers should be brought to bear upon the proper authorities, improvements in standard boiler laws would be obtained.

40 *Electric Equipments.* The uniformity of all laws of this character is essential both to the builder and the buyer, and we again urge that steps be taken by the Society as a body, to further reforms in this direction.

41 *Safety.* Your Committee urges that much can be done by providing better protection of human life, and deplores the great number of accidents that have been caused in connection with the operation of hoisting and conveying machinery. The chief reason for neglecting this phase of the art is that purchasers of hoisting and conveying machinery have not been willing to recognize such additions and refinements where such have added to the cost of the machines. It urges that the question of safety to human life be conscientiously considered by manfacturers, engineers and purchasers.

42 Your Committee recommends that ladders be abandoned, where possible, and stairways with hand rails substituted; that all platforms and walks be provided with double hand rails, toe boards, etc. Where ladders are unavoidable they should be enclosed by lattice work or bars to prevent employees from falling in case of losing hand or foot holds.

43 *Foundations.* Much blame has been put upon builders of this heavy hoisting and conveying machinery on account of inadequate

foundations. Builders of hoisting and conveying machinery seldom design the foundations for their machinery. The design of foundations for big installations comes within a field of foundation specialists who, if employed by purchasers, would obviate many difficulties which might arise.

44 Many docks and storage areas bordering on rivers and lakes, designed for carrying large quantities of bulk materials, have not received the attention in this particular that they should. In many cases expensive concrete dock walls and piling have been pushed out into the rivers by the pressure of the loads carried on the area in the rear. This is particularly true in stock yards around the Great Lakes.

45 Hoisting and conveying machinery should be designed to take care of a reasonable amount of distortion, but it is not reasonable to expect that these massive structures, many of them weighing over a million pounds, can have an unlimited elasticity as to distance between supporting rails. Disregard of suitable foundations results in broken wheel flangers, distorted structures and increased power for propulsion.

46 *Depreciation.* Your Committee finds that some of the largest and most expensive hoisting and conveying machines installed some ten or twelve years ago, are today showing signs of weakness. This indicates that the manufacturers and designers did not anticipate the use (and abuse) that this class of machinery was to receive. Many machines have been designed, sold and guaranteed for a certain capacity, which later developed capacities nearly double that originally contemplated, and carried loads far in excess of those originally estimated. Users do not hesitate to increase the weights of buckets and loads to such an extent that factors of safety are dangerously reduced.

47 Such use helps to depreciate the machines and endangers human life. Purchasers frequently fail to appreciate the great damage to structures caused by the passage of loads at high speed from rail end to rail end. Well designed and efficient machinery, having due regard to human safety and the reduction of costs of upkeep, will many times repay the purchaser.

48 *Wheels.* Your Committee finds that the data concerning track wheels for heavy hoisting and conveying machinery are most limited, both for rolled steel or chilled cast iron wheels. The Committee invites papers bearing on this important subject.

49 *Brakes.* In many hoisting and conveying machines in the past, working to their utmost capacity, the brakes showed need for improvement. Some showed inadequate provisions for the dissipation of the heat developed, others were too short lived. It is no small undertaking to lower a 20-ton load at the rate of 350 ft. per min. once every minute. The dynamic brake has recently taken a prominent place in the control of loads.

50 *Wire Rope.* Your Committee urges the importance of the selection of wire ropes and the design of structures to obtain the greatest life from ropes in service. The transportation of heavy loads together with the limited space for the installation of the operating machinery produces elements which are most destructive to the wire ropes. Ropes become an expensive factor in the operation of machinery of this class. A valuable paper on this subject was recently read before the Institution of Mechanical Engineers of Great Britain, and a digest of it should be before designing engineers. The durability of wire rope is of great importance to all users of hoisting and conveying machinery, and any paper treating exhaustively of this subject would be welcome. Such a paper should bring out the relationships between the diameter of the rope and the sheave over which it is to travel, the diameter of the individual wires in the rope, as well as the relationship between the diameter of the rope and the angularity of the bend when working over a sheave. It would be most desirable to obtain proper specifications for wire ropes for use in hoisting machinery and suitable tests to establish its values.

51 *Ethics.* Your Committee urges that the drawing plans and specifications prepared by any engineering concern are the property of the concern producing them, and holds that buyers should receive such in confidence, decline to exhibit them to others, refuse to permit them to be copied or used by any (but the owner) for purposes of construction.

52 It recommends among engineers that a uniform clause be embodied in all proposals requiring the return of blueprints, drawings and specifications to unsuccessful bidders, and furthermore, that all information submitted for consideration be regarded as the property of the manufacturers' and be treated in confidence.

53 Pirating of ideas is regarded by the Committee as dishonest, and in the end works to the detriment of the buyer as well as to the engineering firms, for it is clear that the most efficient machines are

likely to be produced by the mind that created them rather than by the manufacturer animated purely by the desire of pecuniary profit.

54 The history of nearly every important improvement in the development of the art of hoisting and conveying shows that the engineer freely laid his plans and ideas before the prospective user, both being masters in their own particular field. In order to develop what is required by the producer, complete coöperation between the two is of the utmost importance.

55 Your Committee recommends to all the departments of the United States Government that in asking for proposals, they insert in the invitation that the drawings and blue prints, and all data of a confidential nature shall be retained in confidence and returned to unsuccessful bidders.

Respectfully submitted,

R. B. Sheridan, *Chmn.*
C. K. Baldwin
Alex. C. Brown
O. G. Dale
P. J. Fickinger
F. E. Hulett
Spencer Miller
A. L. Roberts
Harry Sawyer

*Sub-Committee on Hoisting and Conveying*

## DISCUSSION

Spencer Miller in presenting the report for criticism asked if it were too voluminous; should it be further condensed? Should it be elaborated? Should it be illustrated? He said the Committee desired to be of real service.

William Kent said he desired to see the report extended and illustrated, and that the Committee should take one year or two years, if necessary, to do it. The monorail conveyor should be considered and facts regarding it incorporated in the report: what were the limitations of the monorail conveyor, where it should be used, and what were the economical loads. Illustrations showing actual installations should be given, with sizes of bearings, troubles from lubrication, friction, etc. Sketches of the various kinds of monorail conveyors should be shown with the defects of each. This applied to the belt conveyor also.

The Committee should then take up the problems yet to be solved

in conveying, such as getting packages out of the hold of a steamer and putting them into a warehouse in West Street, or the Bush Terminal, or some other place. That whole problem was now being handled in the most crude manner, and if the Committee would take up the matter, it would perform a great service for commerce and engineering.

D. M. Myers was pleased to see the reference in the report to the matter of uniform boiler inspection, and hoped that everything possible would be done by the Society to forward safety in boiler manufacture and inspection.

Spencer Miller was obliged for Mr. Kent's suggestions as to the field in which the Committee could do further work, and said it would go as far as it could.

No. 1404

# DYNAMIC BRAKING FOR COAL AND ORE HANDLING MACHINERY

By Clark T. Henderson, Milwaukee, Wis.

Member of the Society

To many it may seem that dynamic brake control in connection with coal and ore handling machinery is so well known and extensively employed as to call for no further comment unless the commentor is in a position to state new facts or introduce new principles. The writer is, however, of the opinion that many will be interested in a discussion of the advantages and limitations of this method of braking, for in his own experience he has seen several misapplications, resulting either in failure or in disappointment from results obtained.

2 The knowledge that a direct-current electric motor can be operated as a generator and may therefore be used as a retarding device, as well as a driving means, is as old as the motor itself, and at first glance it appears strange that dynamic braking has not come into vogue until recent years. There are, in the writer's opinion, reasons for this which become apparent when the history of the art is considered. The earliest forms of coal and ore handling machines were steam bridges or unloading towers, equipped with small capacity non-automatic tubs. In most cases these were lowered, with the hoisting engines acting as retarding devices. In cases where buckets were retarded in lowering by some sort of friction device no great amount of trouble was experienced, because there was not much energy to be dissipated. With the introduction of the automatic grab bucket the difficulties experienced with mechanical lowering brakes began to multiply, for the automatic buckets were naturally much heavier in proportion to the load handled than the small tubs which they replaced, and the duty imposed upon the lowering brakes was, therefore, greatly increased.

3 After the adoption of the automatic grab bucket the tendency was towards electrically operated apparatus, on which was used the

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

retarding brake, already developed for steam plants. At first these sufficed and were considered satisfactory, but as bucket capacities increased from year to year, and the energy to be dissipated by the lowering brakes increased correspondingly, they began to show unmistakable signs of lack of ability to handle the duty imposed.

4 In analyzing the conditions above outlined, the underlying reasons are quite apparent: In small capacities the unit pressures on friction faces can be low, hence the wear only nominal, and at the same time the radiating surface per watt dissipated can be sufficiently great to insure low temperature rise of parts. In large capacities, however, it becomes practically impossible to retain these proportions, on account of the apparatus becoming unwieldy. As a further disadvantage, the physical effort required to handle large friction lowering brakes is quite considerable, and it is axiomatic among designers of this class of machinery that prolonged high speed is not possible if the operator is called upon to exert any great effort.

5 Several builders brought out equipments on which brakes were controlled by air or hydraulic cylinders, and only pilot valves handled by the operator. These arrangements did not overcome heating difficulties, and in many cases were found unsatisfactory because of the lack of sensitiveness of control. When the operator handles friction brakes directly, he can, by sense of touch, determine when they are dragging to a proper degree, but when operated through a pilot valve, this sense is lost. It was natural, therefore, that the first 10-ton unloading machines to be electrified should have dynamic braking employed to retard the descending bucket. Manufacturers of bridges next adopted this form of control, first for ore handling and later for coal. It is now used extensively for ordinary traveling cranes, and from the present outlook it is quite possible that the mechanical lowering brake will be entirely obsolete within a few years.

6 Some form of holding brake must be employed in connection with dynamic braking, for while the motor acting as a generator can prevent the load from descending faster than a few inches per minute, it can never entirely stop the load, except through the aid of a mechanical holding device, which in some cases is electrically operated and automatic; in others manually controlled and non-automatic.

7 On the earliest installations compound wound direct-current motors were almost invariably employed, the shunt winding being deemed necessary to insure prompt building up of the fields. Later developments showed that series motors, the characteristics of which

made them so desirable for hoisting, could be used equally well if their fields were temporarily excited just at the moment of releasing the holding brake and the excitation current cut off as soon as the machine built up as a generator. The cutting off of the excitation current is desirable, not only from the standpoint of current economy, but also from that of motor heating, for whatever exciting current is passed through the motor fields simply adds to the general heating of the machine. The use of series motors is also desirable from the standpoint of simplicity. Where compound motors are employed it is considered the best practice to leave the shunt field continuously excited and the heating consequent to the continuous excitation of these shunt windings is considerable.

8 One of the difficulties first encountered in connection with dynamic brake control systems was that the operator, who must necessarily be able to reverse his hoist motors in order to open his grab quickly and to start down without loss of time, would in many cases leave the power on almost all the way down and would in consequence race the hoist motor to such an extent as to throw out commutator bars and tear armature coils from their fastenings. Even if these things did not occur, the commutation at these high speeds was sure to be very poor, and motors were apt to flash over when the operator threw his controller back from the reverse power to the dynamic brake position. It can readily be seen that the closure of the dynamic braking circuit at a time when the armature speed was several times more than normal would cause the flow of a braking current of excessive value.

9 These difficulties led to the adoption of a control scheme which permits the operator to reverse the hoist motors when opening buckets and when starting to lower, and which causes the machine to operate as a motor until a predetermined speed is reached, when it automatically and without interruption of the circuit becomes a generator. This system was further perfected by the introduction of current relays in the braking circuit, with which the current of retardation can be limited to a reasonable value, and a value well within the commutation limits of the motor. As a further refinement control systems have been developed for those installations where the holding brake is automatically operated which insure the maintenance of the brake in the released position until the load has come substantially to rest.

10 Some plants installed during the development stage have

not been as satisfactory as the purchasers had hoped. Against the later and perfected installations only two indictments can properly be brought: (*a*) motors sometimes overheat; (*b*) machines have insufficient lowering speed.

11 Assuming the installation of a proper control system, the overheating of motors can invariably be traced to improper selection. Hoist motors cannot properly be selected on the basis of horsepower to be developed during the hoisting period. The service for which they are employed is so intermittent as to necessitate their selection by what is known as the root-mean-square method, or its equivalent. Generally speaking, motors required for a given hoist operating on a given cycle must have a thermal capacity which is greater by $33^1/_3$ per cent if dynamic braking is employed than will be required if the lowering of the bucket is accomplished by mechanical brakes.

12 Lowering speeds, possible with standard motors and dynamic brake control, never seem to be criticised except in connection with high-speed unloading plants. Generally speaking, it is not good practice to figure on obtaining lowering speeds which are greater than 200 per cent of the full load hoisting speed, unless special motors are employed, and it is a mighty good plan to figure on holding the lowering speeds down to one and one-half times full load hoisting speed if possible. The commutation of standard series wound motors of what is called the mill type (and this is the type which is generally used on coal or ore handling bridges) is not good if they are operated above 200 per cent of full load speed; furthermore the energy stored in their armatures becomes too great for rapid control when they are operated above this limit. If dynamic braking is to be a complete success on fast plants, it will be necessary for the designers of that machinery to select special slow-speed motors of the interpole type. There is little or no doubt but that with properly selected motors lowering speeds can be obtained with dynamic brake control which are just as high as those which are possible with mechanical lowering brakes. The apparatus will, however, be somewhat more expensive, and may be commercially impossible for that reason.

13 So far we have considered dynamic braking only as applied to direct-current motors used for bridge hoist service. In some cases this form of control has been used in connection with trolley traverse motors, but for this service it is not nearly so well adapted; in the writer's estimation its fitness is questionable. In hoist service there is always a definite force (that of gravitation acting on the bucket)

tending to run the hoist motor as a generator at all times when braking is required. On trolley traverse service the force driving the motor is exceedingly variable, being that of trolley inertia, and therefore proportional to the square of the trolley speed.

14 In hoist service dynamic braking is never required when the motor is rotating in the hoisting direction, and the reversal of the direction of rotation, before the motor is called upon to act as a generator, makes it possible for a series machine to be self-exciting without reversing the relation of armature and field, as is necessary on a trolley traverse motor when braking is to be accomplished while the motor continues to revolve in a given direction. The result is that a dynamic brake hoist controller is a much simpler and much more reliable piece of apparatus than a corresponding control for a trolley traverse motor.

15 The variable amount of energy to be absorbed on trolley control service and the comparatively small number of steps commercially possible make it impractical to stop by dynamic braking without making the bucket swing. Air braking, under the control of a standard railway type brake valve, providing, as it does, an infinite number of gradations of braking, is, in the writer's opinion, preferable for heavy duty, and the manually applied brake preferable for light service.

16 When a motor runs, say in a clockwise direction, it will be found that the best results will be obtained by setting the brushes a little ahead of the neutral point and thus giving it what is called the "motor lead." What is motor lead for one direction of rotation is "generator lead" for the opposite direction of rotation. Hoist motors can have their brushes given motor lead for hoisting and this will aid them to commutate properly while acting as generators during lowering. Any reverse power that they may be called upon to develop will be so small in comparison with that required in hoisting that their generator lead for the lowering direction will not be found objectionable. The trolley traverse motors when used with dynamic brake control may be called upon to act either as generators or motors in both directions, and therefore cannot be thus aided. For this reason it is especially desirable to employ interpole motors when dynamic braking is used for trolley traverse control.

17 In connection with car dumping machines there is absolutely no doubt but that dynamic brake control for the tipple motors is highly advisable, and so far the writer has never heard any valid

objections made to its employment for this service. The lowering speeds demanded are never excessively high—one and one-half times full load hoisting speed invariably seeming to be considered quite satisfactory.

18 Dynamic brake control systems are sometimes employed in connection with car haul for pulling cars upon a dumper. In those cases where it is employed it has proved quite satisfactory. Braking is not ordinarily required when moving the mule car up the incline, and such braking as may be required in lowering is so small in comparison with the capacity of the motors necessarily employed as to make the problem an exceedingly simple one.

19 Of late years there has been a considerable number of bridges equipped with alternating-current motors and in some instances dynamic brake control has been employed in connection with those motors operating the hoists. The writer has not heard of any installations where this form of control is utilized in connection with trolley traverse motors.

20 A three-phase alternating-current motor of the slip ring type has to a large extent the characteristics of a direct-current shunt wound machine. Having these characteristics, it resists any force which tends to make it revolve above synchronous speed. A motor whose synchronous speed is 750 r.p.m., for example, will operate at about 750 r.p.m. when exerting full load torque. If a force equal to full load torque tends to revolve this motor above synchronous speed, the machine will develop a retarding force equal to the propelling force at approximately 780 r.p.m. On account of this characteristic, three-phase motors can be connected directly to the line for reverse operation and will prevent the lowering speed from exceeding the full load hoisting speed more than a few per cent. The motors cannot, however, be used to slow down the load, for only this one braking speed is available. On some installations attempts have been made to obtain perfect braking control by the excitation of the stator winding of the hoist motors from a source of direct-current supply. When the stator windings are thus excited the motors act as generators and the currents set up in the motor windings serve to retard the descending load. It has been found, however, that to bring the descending bucket substantially to rest, a very heavy exciting current is required in the stator winding. In fact, the exciting current required is so high that if prolonged for any considerable period of time, it will cause the winding to overheat. It has been necessary, therefore, to incorporate with these alternating-current dynamic brak-

ing control systems adapted to give graduated control, means for varying the stator excitation in proportion to the braking effort which the motor is expected to exert. There is one successful installation of this character with which the writer is familiar, but the maximum lowering speed it is possible to obtain is only one and one-half times full load hoisting speed. Above this speed the machine develops a tendency to run away. Then again, the machines are much slower in operation because the fly-wheel effect of the rotors in alternating-current motors is considerably greater than that of armatures in direct-current motors of corresponding capacity.

21 In conclusion, it would seem proper to state that dynamic brake control with properly selected motors is desirable on the hoist motion of all direct-current coal and ore handling machinery; that the use of dynamic brake control in connection with trolley traverse motors is undesirable on both direct current and alternating current, and that the use of dynamic brake control is desirable in connection with alternating-current hoist motors, provided there is a variable excitation of the stator windings during the braking period.

# SYMPOSIUM ON STEEL FRAME BOX CARS

## No. 1405 *a*

## STEEL UNDERFRAME BOX CARS

By George W. Rink, Jersey City, N. J.

Member of the Society

In approaching a subject of such importance to railroads as steel underframe box cars, it is surprising, in view of the interchange of such cars among the railroads, that more has not been accomplished during the past five years toward standardization in design of the various component parts, particularly those which affect the cost of maintenance and require constant repairs due to wear and unavoidable accidents.

2 During the year 1912, there were built 107,887 box cars of various capacities and dimensions, all varying vastly in detail design of important parts which require frequent renewal, thus making it necessary for all railway storehouses to carry an unnecessarily large stock of repair parts running into very large sums of money. Standards have been adopted by the Master Car Builders' Association which have in a large measure reduced the amount of stock necessary to be carried. I believe the time has arrived to introduce additional standards affecting the maintenance of box cars which can also be applied to all types of freight cars used in interstate business.

3 It is reasonable to assume that every railroad manager desires to purchase cars built in a substantial manner. In the absence of standard construction and because of competition, the car builders, when asked to furnish estimates and designs, will sometimes figure on material too light for the service. This, however, is not the fault of the car builder, since, from my own experience, I know that they will gladly add the material where needed, provided they are paid a fair price for the car.

4 From my observation of steel underframe box cars, I must

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

conclude that engineers did not understand the importance of low fiber stresses in the early designs of steel underframes; sufficient attention was not given to the tremendous impact blow which the center sills and car framing found it necessary to resist, with the result that large sums of money are now being spent by railroads in making repairs to these cars by reinforcing broken center and draft sills, applying larger capacity draw gears and attachments, and heavier sills. It is not alone the use of the larger locomotives today which has called for a more thorough investigation of the subject of car design and construction, but also the severe shocks which cars are receiving in classification yards. Also the superstructure of box cars should receive just as much attention as the underframe, for how can the roofs be kept in alignment on cars having wood side posts and braces and loose tie rods. The roof is bound to work loose, resulting in leaks which prevent the use of the car for certain commodities.

5 The application of steel underframes to old cars will no doubt prolong their lives. This is now being done by almost every railroad on cars built just prior to the advent of all-steel underframe box cars, but care should be taken, to see that sufficient metal is provided to withstand the present service requirements, keeping in view a margin of safety for the future, as no doubt it would be desirable to maintain in service for at least ten years cars to which these steel center sills were applied. The reinforcement of the ends of box cars has received attention by the Car Construction Committee of the Master Car Builders' Association, and they have presented recommendations in the way of designs for old and new box cars. This committee likewise made recommendations with a view to establishing minimum area of center sills to resist end strain.

6 Railway officials in charge of car repairs have seen the results of poor designing and light construction of the earlier steel underframe cars, and during the past three years have materially assisted in the development of the art by insisting upon the production of a stronger car, one that will hold together in all kinds of service with the minimum cost of repairs.

7 I have compiled certain relative data from designs of eight wooden upper frame cars and one steel frame car, all of which have inside lining and outside sheathing, also six steel upper frame cars with only inside lining. All of these cars represent the latest designs of box cars now being built or recently placed in service by various railroads. The data obtained have been divided into the following

groups, special cars for automobiles and furniture having been disregarded:

Table 1 General dimensions of box cars with outside sheathing
Table 2 General dimensions of box cars with inside sheathing
Table 3 Dimensions of wood side framing
Table 4 Dimensions of steel side framing
Table 5 Dimensions of end framing for cars with wood side frames
Table 6 Dimensions of end framing for cars with steel side frames
Table 7 Dimensions of side door openings, size of door and distance from rail to top of floor
Table 8 Dimensions of running boards, type of roof and car line, width and height at eaves, etc.
Table 9 Distance between body bolsters, spread of truck wheels, etc.
Table 10 Dimensions of draft sills, side sills, floor supports, etc.
Table 11 Area, section moduli, etc., of steel underframes at center for cars with wood side frames
Table 12 Area, section moduli, etc., of steel underframes near bolster for cars with wood side frames
Table 13 Area, section moduli, etc., of steel underframes at center for cars with steel side frames
Table 14 Area, section moduli, etc., of steel underframes near bolster for cars with steel side frames

8 *Table* 1. By referring to this table, it will be noted that the inside dimensions, such as length and width for 36-ft. box cars are the same with but one exception—the inside height varies considerably, due to the type of the carline used. The American Railway Association adopted on October 23, 1901, standard inside dimensions for box cars as follows: length, 36 ft.; width, 8 ft. 6 in.; height, 8 ft. No recommendations were made by the Master Car Builders' Association, as requested by the American Railway Association, to establish external dimensions for the 36-ft. box car. By referring to the table it will be noted that the length over sheathing varies from 36 ft. 10¼ in. to 37 ft. 1⅛ in., with length over striking plates varying from 37 ft. 7⅝ in. to 38 ft. 8⅞ in., the variation being accounted for by the use of different types of end construction. No doubt these differences will continue on future designs of cars until a standard end frame and end sill construction is made compulsory.

9 *Table* 2. This shows similar dimensions for types of steel side frame cars, and it would appear that the inside dimensions, especially of the long cars, are made to suit the whims of various designers. There is really no excuse for such a condition, as it means a further drifting away from the car of standard dimensions which should by reason of duplication in all its parts, facilitate repairs, bearing in mind that it is going to be a more costly proposition for railroads to repair, and at times replace, parts of damaged superstructures of steel frame cars than when made of wood. It appears

TABLE 1 GENERAL DIMENSIONS OF BOX CARS WITH OUTSIDE SHEATHING

| Road | Capacit | Light Weight of Car | Inside Dimensions | | | Length Over | | | Width Over Side Sills |
|---|---|---|---|---|---|---|---|---|---|
| | | | Length | Width | Height | Striking Plate | End Sill | Sheathing | |
| D. L. & W. | 60,000 | 37,300 | 36′ 0′ | 8′ 6″ | 8′ 0″ | 37′ 7⅝″ | *36′ 8⅝″ | 36′ 10¼″ | 9′ 0½″ |
| C. R. R. of N. J. | 60,000 | 39,000 | 36′ 0″ | 8′ 6″ | 8′ 0″ | 38′ 3½″ | 37′ 6½″ | 37′ 1⅛″ | 9′ 1″ |
| P. & L. E. | 80,000 | 39,600 | 36′ 0″ | 8′ 6″ | 8′ 4″ | 38′ 0½″ | 36′ 10¾″ | 37′ 0⅜″ | 9′ 2″ |
| N. Y. O. & W. | 80,000 | 40,000 | 36′ 0″ | 8′ 6″ | 8′ 2½″ | 38′ 8⅞″ | 37′ 10⅞″ | 36′ 11¾″ | 9′ 1⅝″ |
| P. & R. | 80,000 | 48,100 | 36′ 2″ | 8′ 7″ | 8′ 2⅛″ | 38′ 2½″ | *37′ 0″ | 37′ 1¼″ | 9′ 2″ |
| B. & Albany | 80,000 | 39,800 | 36′ 0″ | 8′ 6″ | 8′ 4″ | 38′ 1″ | 36′ 10¾″ | 37′ 0⅜″ | 9′ 2″ |
| C. & N. W. | 80,000 | 41,200 | 40′ 0″ | 8′ 5¼″ | 7′ 11½″ | 41′ 10⅞″ | 40′ 9½″ | 40′ 11⅛″ | 9′ 1¼″ |
| Un. Pac. | 100,000 | 42,700 | 40′ 0″ | 9′ 1¾″ | 9′ 2¼″ | 42′ 1¼″ | *40′ 8⅜″ | 40′ 9⅞″ | 9′ 8½″ |
| C. & O. | 80,000 | 39,700 | 36′ 0″ | 8′ 6″ | 8′ 0″ | 38′ 6½″ | *37′ 10½″ | 36′ 11¼″ | 9′ 1⅝″ |

* Between End Sills.

TABLE 2 GENERAL DIMENSIONS OF BOX CARS WITH INSIDE SHEATHING

| Road | Capacity | Light Weight of Car | Inside Dimensions | | | Length Over | | | Width over Side Sills |
|---|---|---|---|---|---|---|---|---|---|
| | | | Length | Width | Height | Striking Plate | End Sill | End Posts | |
| Erie | 80,000 | 38,800 | 36′ 0″ | 8′ 6½″ | 8′ 0½″ | 38′ 1¼″ | *36′ 11½″ | 36′ 11½″ | 8′ 9½″ |
| Can. Pac. | 80,000 | 38,300 | 36′ 0″ | 8′ 6½″ | 8′ 0½″ | 38′ 1¼″ | *36′ 11½″ | 36′ 11½″ | 8′ 9½″ |
| Wabash | 80,000 | 36,200 | 36′ 5⅝″ | 8′ 6″ | 8′ 0″ | 38′ 5⅝″ | *37′ 4⅞″ | 37′ 4⅞″ | 8′ 9¼″ |
| Frisco | 80,000 | 40,700 | 40′ 0″ | 8′ 6″ | 8′ 0″ | 42′ 0¾″ | *40′ 11″ | 40′ 11″ | 8′ 8¼″ |
| Un. Pac. | 100,000 | 41,800 | 40′ 5⅜″ | 9′ 2″ | 9′ 0″ | 42′ 1¼″ | 40′ 8⅜″ | 41′ 4⅜″ | 9′ 5″ |
| P. R. R. | 100,000 | 47,900 | 40′ 6″ | 8′ 10″ | 8′ 0″ | 42′ 6″ | 42′ 6″ | 41′ 8½″ | 9′ 0½″ |

* Between End Sills.

that there was a demand for larger cars than those adopted by the American Railway Association. At a meeting of this body held November 20, 1912, the Committee on Standard Dimensions for Box Cars, after ascertaining the clearances of various railroads, decided that a box car 40 ft. 6 in. long, 8 ft. 6 in. wide and 9 ft. high was the largest car which could be constructed with due regard for clearances. I believe it undesirable to increase the inside height of the car to 9 ft. on account of increasing the height of the center of gravity of the car from the top of the rail. The present inside cross dimensions should be maintained, and the car should simply be lengthened to 40 ft. in order to obtain increased cubic capacity. Attention is also called to the great variations in weight of cars shown in Tables 1 and 2, which leads me to believe that either some of the cars are built too light or others are carrying excess weight in the way of material not properly distributed.

10 *Table* 3. This table is very interesting in pointing out what various designers consider necessary in the way of sizes of material for posts and braces which make up the wooden body frame, the general arrangement of which is shown in Figs. 1 to 4. In 1904, the style of framing recommended by the Master Car Builders' Association, and shown in Fig. 2, was adopted for cars of 60,000 lb. capacity, and that shown in Fig. 3, for cars of 80,000 and 100,000 lb. capacity. There is this one exception, however, that the Master Car Builders' design indicates the use of one belt rail, whereas, it is now necessary to provide two belt rails in order to secure properly the outside sheathing. This extra rail also adds materially to the rigidity of the side frame, assisting it against end strains.

11 *Table* 4. In this table is shown a comparison on sizes of material used in different members of the steel frame box car. This type of framing, as applied to a large number of box cars recently constructed, lends itself very readily to standard construction; and it is well to note that, with the exception of intermediate posts and braces, we are again confronted with a variety of sections and weights of materials comprising the side plate, side sill, corner and door posts. There is no good reason why this condition should continue; these cars will be in service practically all over the country, and barring wrecks, the frames will be subject to the same stress due to impact blows, car lading and weight of superstructure.

12 The general designs of steel side frames are shown in Figs. 5 to 12. It will be noted that standard structural material has been used throughout, with the exception of the Pennsylvania Railroad box

TABLE 3 DIMENSIONS OF BODY SIDE FRAMING—WOOD CONSTRUCTION

| Road | Capacity | Length over Sheathing | Side Posts | Side Braces | Side Plates | Corner Posts | Side Door Posts | Belt Rail | | Style of Frame |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | Size | No. | |
| D. L. & W. | 60,000 | 36′ 10¼″ | 5″x2⅜″<br>with<br>⅜″x2¼″ Pl. | 5″x2⅜″ | 6¼″x3¼″ | 5″x5″<br>with<br>3″x3″x5/16″∟ | 5″x4¾″ | 4½″x2⅜″ | 2 | Fig. 3 |
| C. R. R. of N. J. | 60,000 | 37′ 1⅛″ | 5″x2½″ | 5″x2½″ | 7½″x3″ | 4″—7½ lb. I<br>4″x5½″wood<br>2⅜″x4″filler | 4½″x4⅜″ | 4″x3½″ | 2 | Fig. 2 |
| P. & L. E. | 80,000 | 37′ 0⅜″ | 5″x3″ | 5″x3″ | 6″x4″ | 5½″x5½″ | 5½″x5 1/16″ | 4½″x3″ | 2 | Fig. 1 |
| N. Y. O. & W. | 80,000 | 36′ 11¾″ | 5″x3″<br>and<br>3″—4 lb. [ | 5″x3″ | 7½″x4″ | 5″x5″ | 5½″x5″<br>and<br>3″—4 lb. [ | 3½″x3″ | 2 | Fig. 2a |
| P. & R. | 80,000 | 37′ 1¼″ | 6″x3⅜″ | 6″x2 5/16″ | 8″x3″ | 7¼″x7″ | 6″x4⅞″ | 6″x3⅜″ | 2 | Fig. 2 |
| B. & Albany | 80,000 | 37′ 0⅜″ | 5″x3″ | 5″x3″ | 6¾″x4″ | 5½″x5½″ | 5½″x5½″ | 4½″x3″ | 2 | Fig. 4 |
| C. & N. W. | 80,000 | 40′ 11⅛″ | 5″x3¾″ | 5″x2¾″ | 6″x3″ | 6″x5½″ | 5″x4 11/16″ | 4″x2¾″ | 1 | Fig. 3 |
| Un. Pac. | 100,000 | 40′ 9⅞″ | 5″x2½″<br>and<br>½″x2¼″ Pl. | 5″x2½″ | 7″x3½″ | 5″x5″ | 6″x4¾″<br>and<br>½″x4½″ Pl. | 4″x2½″ | 2 | Fig. 3a |

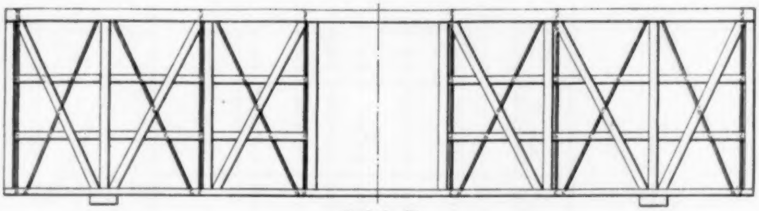



FIG. 1 SIDE FRAME USED ON P. & L. E. CARS SHOWING WOOD POSTS AND BRACES WITH DIAGONAL ROD BRACING

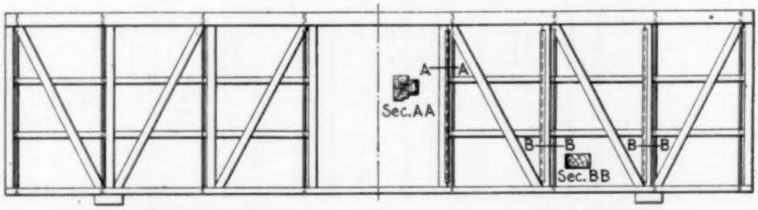



FIG. 2 FIG. 2a

FIG. 2 SIDE FRAME USED ON P. & R. AND C. R. R. OF N. J. CARS WITH WOOD POSTS AND BRACES

FIG. 2a SIDE FRAME USED ON N. Y. O. & W. CARS WITH WOOD POSTS AND BRACES; POSTS REINFORCED BY CHANNELS

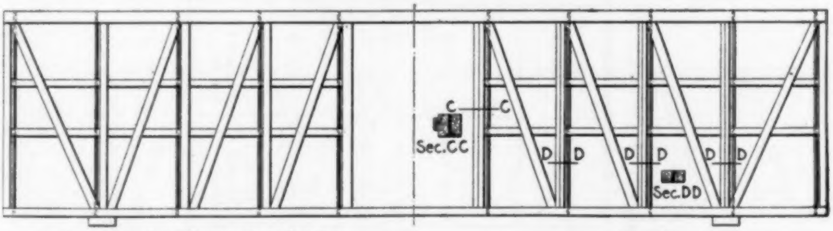



FIG. 3 FIG. 3a

FIG. 3 SIDE FRAME USED ON C. & N. W. AND D. L. & W. CARS HAVING NINE PANELS WITH WOOD POSTS AND BRACES. TYPE OF FRAME ADOPTED BY THE M. C. B. ASSOCIATION FOR CARS OF 80,000 AND 100,000 LB. CAPACITY

FIG. 3a SIDE FRAME USED ON UN. PAC. CARS HAVING NINE PANELS WITH WOOD POSTS AND BRACES; POSTS REINFORCED BY PLATE

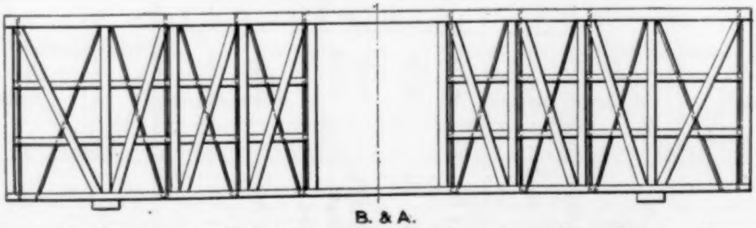



FIG. 4 SIDE FRAME USED ON B. & A. CARS HAVING NINE PANELS WITH WOOD POSTS AND DIAGONAL ROD BRACING

TABLE 4 DIMENSIONS OF BODY SIDE FRAMING—STEEL CONSTRUCTION

| Road | Capacity | Side Sills | Length over End Posts | Side Posts | Side Braces | Side Plates | Corner Posts | Side Door Posts | Inside Sheathing | Size of Sheathing | Style of Frame |
|---|---|---|---|---|---|---|---|---|---|---|---|
| C. & O. | 80,000 | 8"—11¼ lb. [ 5½"x3" wood | 36' 9⅝" | 3"—6.7 lb. Z | 3"—6.7 lb. Z | 3½"x3"x 5/16"∠ 2½"x5" wood | 4"x3"x 5/16"∠ 5"x4" wood | 3"—6.7 lb. Z 4¾"x3¾" wood | Horizontal | 13/16"x5¼" | Fig. 5 |
| Erie | 80,000 | 8"—11¼ lb. [ flanges in | 36' 11½" | 3"—6.7 lb. Z | 3"—6.7 lb. Z 3 1/16"—8.4 lb. Z | 4"—8.2 lb. Z | 5"x5"x⅜"∠ | 5"x3½"x 5/16"∠ with 3"—4 lb. [ | Horizontal | 1½"x5¼" | Fig. 6 |
| Can. Pac. | 80,000 | 8"—11¼ lb. [ flanges in | 36' 11½" | 3"—6.7 lb. Z | 3"—6.7 lb. Z 3 1/16"—8.4 lb. Z | 4 1/16"—10.3 lb Z | 5"x5"x⅜"∠ | 5"x3½"x 5/16"∠ 3"—4 lb [ | Horizontal | 1½"x5" | Fig. 6 |
| Wabash | 80,000 | 8"—11¼ lb. [ flanges in | 37' 4⅞" | 3"—6.7 lb. Z | 3"—6.7 lb. Z | 3"—6.7 lb. Z | 4"x4"x¼"∠ | 3"—6.7 Z | Horizontal | 1⅝"x5¼" | Fig. 6 |
| Frisco | 80,000 | 5"x3"x⅜"∠ 3"x4⅝" wood | 40' 11" | 3"—6.7 lb. Z | 3"—6.7 lb. Z 3 1/16"—8.4 lb. Z | 4"x3"x 5/16"∠ | 5"x4"x⅜"∠ | 4"—8.2 lb. Z | Horizontal | 1½"x3¼" | Fig. 6a |
| Un. Pac. | 100,000 | 9"—13¼ lb. [ flanges in | 41' 4⅜" | 3"—6.7 lb. Z | 3"—6.7 lb. Z | 4"—8.2 lb. Z | 5"x5"x⅜"∠ | 5"x3½"x 5/16"∠ Pressed U shape | Horizontal | 1½"x5¼" | Fig. 7 |
| P. R. R. | 100,000 | 6"x4"x⅜"∠ | 41' 8½" | ¼" Pl. Pressed U shape | ¼" Pl. Pressed U shape | 4"x6"x⅜"∠ | 4"x4"x⅜"∠ | Bulb Angle | Vertical | 1¼"x3¼" | Fig. 8 |

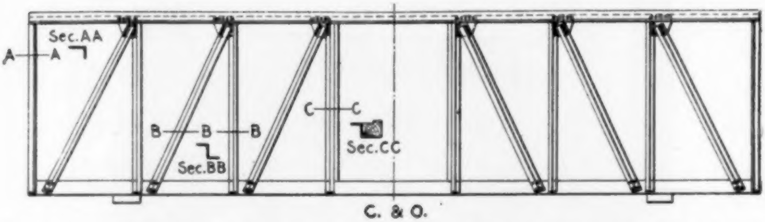



FIG. 5 STEEL SIDE FRAMING USED ON C. & O. CARS HAVING SEVEN PANELS WITH END PANEL BRACED DIAGONALLY; STRUCTURAL MATERIAL USED

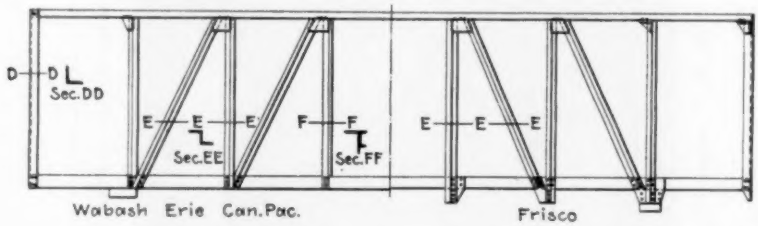



FIG. 6 FIG. 6a

FIG. 6 STEEL SIDE FRAMING USED ON THE WABASH, ERIE AND CAN. PAC. CARS HAVING SEVEN PANELS AND NO END PANEL BRACING; STRUCTURAL MATERIAL USED

FIG. 6A STEEL SIDE FRAMING USED ON FRISCO CARS WITH SEVEN PANELS AND NO END PANEL BRACING; STRUCTURAL MATERIAL USED

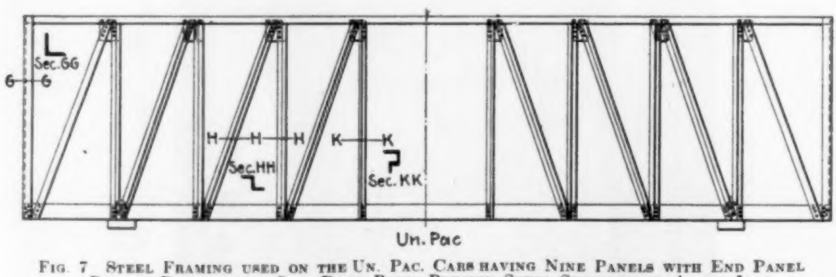



FIG. 7 STEEL FRAMING USED ON THE UN. PAC. CARS HAVING NINE PANELS WITH END PANEL BRACED DIAGONALLY; SIDE DOOR POSTS PRESSED STEEL SECTION AND ANGLE IRON

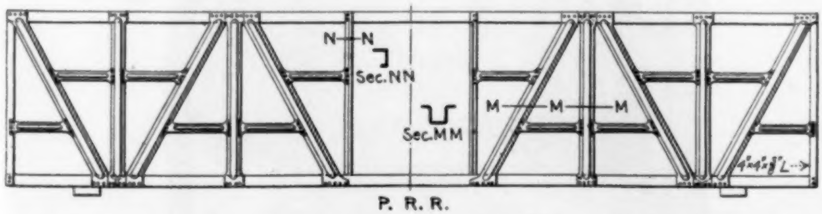



FIG. 8 STEEL SIDE FRAMING USED ON P. R. R. CARS WITH SEVEN PANELS WITH END PANEL BRACING FROM BOLSTER TO SIDE PLATE AT END: PRESSED STEEL U-SECTIONS WITH FLATTENED ENDS FOR SECURING TO SIDE PLATE AND SIDE SILL

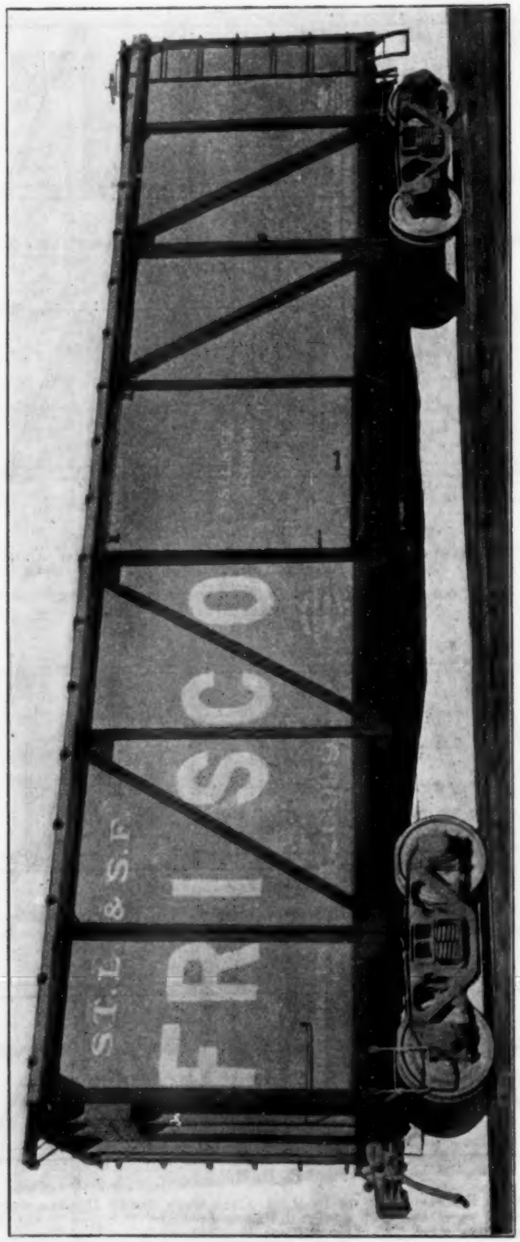



Fig. 9 40-Ton Outside Steel Frame Box Car used on Frisco System

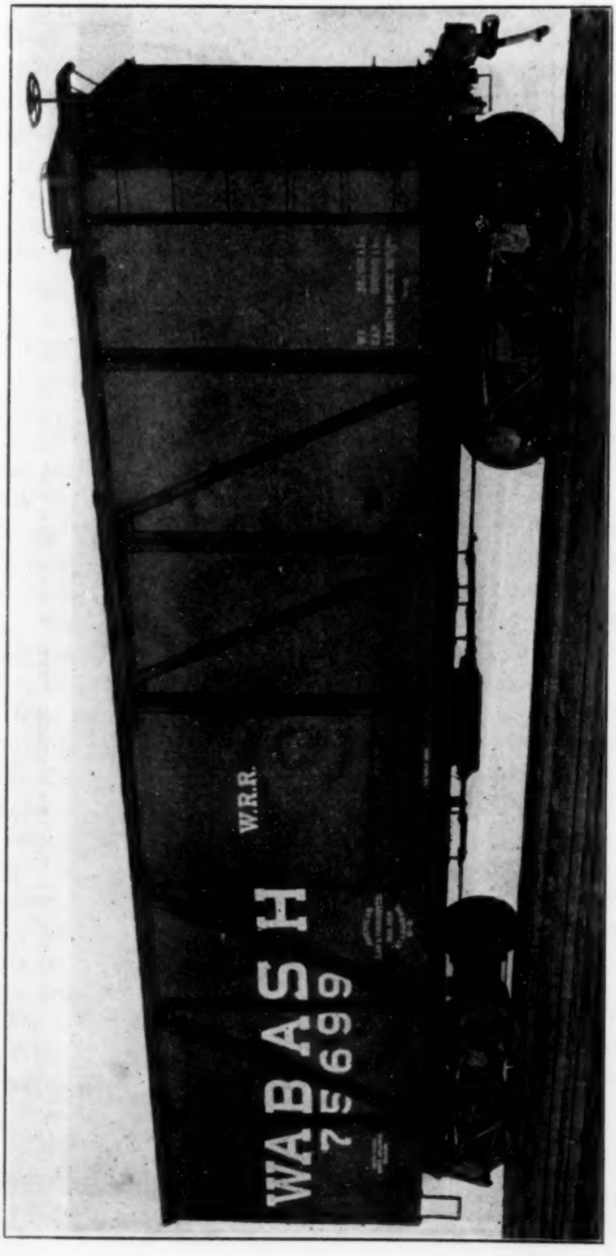



Fig. 10 40-Ton Outside Steel Frame Box Car used on Wabash R. R.

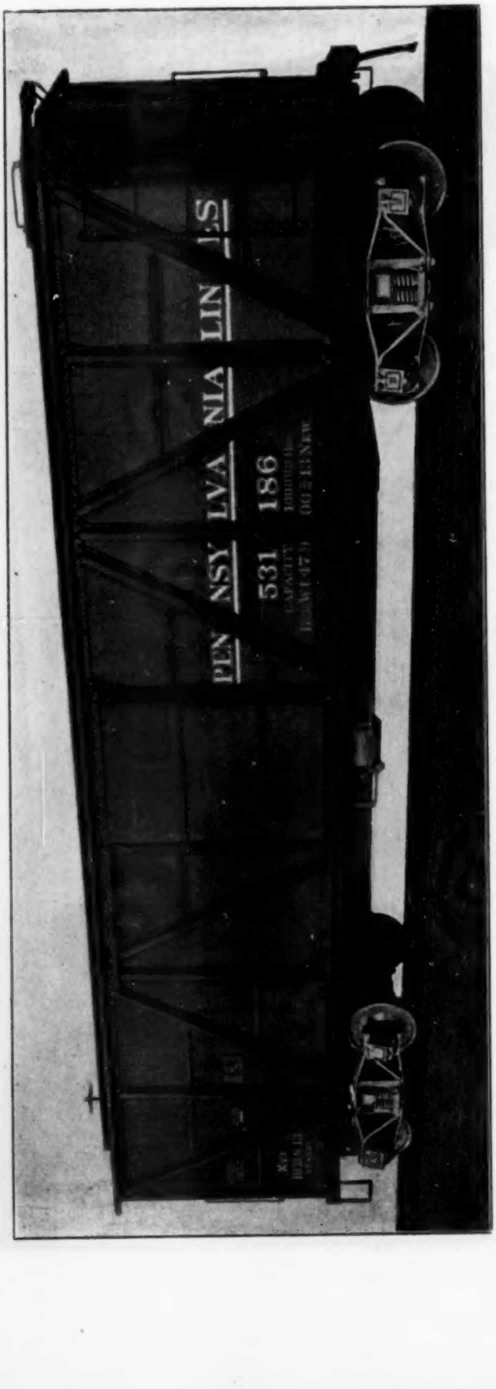



FIG. 11 50-TON OUTSIDE STEEL FRAME BOX CAR USED ON P. R. R.

cars. The design of frame of this railroad, shown in Fig. 8, makes a splendid appearance, is well arranged and provides for vertical inside sheathing. Due to the construction of the posts and braces, slight contact blows from obstructions on neighboring tracks will not affect the appearance of the brace as they will if structural steel is used, which is more liable to distort. However, I am not in favor of pressed steel, on account of its higher first cost as compared with stock rolled steel and also of the fact that the ends of the posts and braces are flattened out where they connect to the side sill and plate, thus weakening them, especially at the bottom, where depth of the member is desirable to resist bending due to shifting of load or side impact blow.

13 I prefer the frame construction shown in Fig. 7 for cars having an inside length of 40 ft. or over and the same construction for cars 36 ft. inside length, omitting one panel from each side. A diagonal brace has been introduced in the end panel and is in tension due to lading, thus relieving the stress on the side sill near the bolster. It also ties the lower corner of the side frame. If the corner post were to receive an end blow directly over the push pole pocket, with the assistance of an end diagonal it would keep the panel square. Several designs (Figs. 6 and 6a) show omission of this brace, but for what reason I do not know, as it could be used to advantage also to secure the inside lining in this panel of wide span.

14 The question of frame design will depend largely on how the sheathing is to be extended, i. e., vertically or horizontally. The sheathing extends horizontally for all designs of steel frames shown, except for the pressed steel frame. I believe vertical sheathing is preferable as it protects the lading against water. In the case of horizontal sheathing, rain can beat in at the joints and the grooves or beading form gutters that have a tendency to deposit the water into the car, both at the door and end posts. This is an important question in the case of cars used for transporting grain. Standard size sheathing should also be adopted. Referring to the last column in Table 4, the thickness varies from $1\frac{1}{4}$ in. to $1\frac{5}{8}$ in. and the width from $3\frac{1}{4}$ in. to $5\frac{1}{4}$ in. It is reasonable to assume that if all roads having cars built of this type were to use a uniform size of material, it would be far easier and cheaper to make repairs.

15 *Table* 5. This shows the various sizes of material used in end frame construction for cars with wood side frames. Various designs have been introduced, which are shown in Figs. 13 to 19. The use of I-beam posts seems to predominate, using wood fillers on each

TABLE 5 DIMENSIONS OF END FRAMING FOR CARS WITH WOOD SIDE FRAMING

| Road | End Sills | End Plates | End Posts | End Braces | Corner Posts | Type of Frame |
|---|---|---|---|---|---|---|
| D. L. & W. | 9″—15 lb. [ flanges out; also 3″x3″x 3/16″∠inside | 3½″x11¾″ | 2—5″x3⅜″ reënforced with 3″x3″x 3/16″∠ | 5″x3⅜″ | 5″x5″ with 3″x3″x 3/16″∠ | Fig. 19 |
| C. R. R. of N. J. | 12″—40 lb. [ flanges in ¼″x10″ Pl. & 3″x2½″x 3/16″∠top | 4″x13½″ | 4″—7½ lb. I with 2—2⅜x4″ fillers | 4″x4″ | 4″—7½ lb. I 4″x5½″ wood 2⅜″x4″ filler | Fig. 16 |
| P. & L. E. | ⅜″x10¾″ Pl. 4″x4½″x⅜″∠ top 3″x3″x⅜″∠s top and bott. inside | 4″x12″ | 4″—7½ lb. I with 2—2″x4″ fillers | 5″x4″ | 5½″x5½″ | Fig. 13 |
| N. Y. O. & W. | 12″—20.5 lb. [ flanges in ¼″x12″ top Pl. 3″x2½″x¼″∠top | 4″x13″ | 4″—7½ lb. I with 1—4″x2½″ wood filler | 4″x4″ | 5″x5″ | Fig. 14 |
| P. & R. | 13″—32 lb. [ flanges out & cast iron filler 3″x2″x 3/16″∠top | 4″x15¼″ | 6″x4″ with 3/16″x6′—1⅜″ Plate | None | 7¼″x7″ | Fig. 17 |
| B. & Albany | ½″x10″ Pl. 5″x3½″x⅜″∠top 3″x3″x⅜″∠s top and bott. inside | 4″x12″ | 4″—7½ lb. I with 2—2 3/16″x4″ fillers | 5″x4″ | 5½″x5½″ | Fig. 13 |
| C. & N. W. | 6″x9¼″ wood with ½″x9″ Plate in rear | 3″x14⅛″ | 5″—9¾ lb. I with 2—3⅛″x5″ fillers | 5″x3″ | 6″x5½″ | Fig. 15 |
| Un. Pac. | 8″—13¾ lb. [ flanges out 3½″x3″x 3/16″∠top inside and 5″x5 3/16″ wood | 3⅜″x13 | 5″x4⅜″ with ½″x3½″ flitch plate | 5″x3⅜″ | 5″x5″ | Fig. 18 |

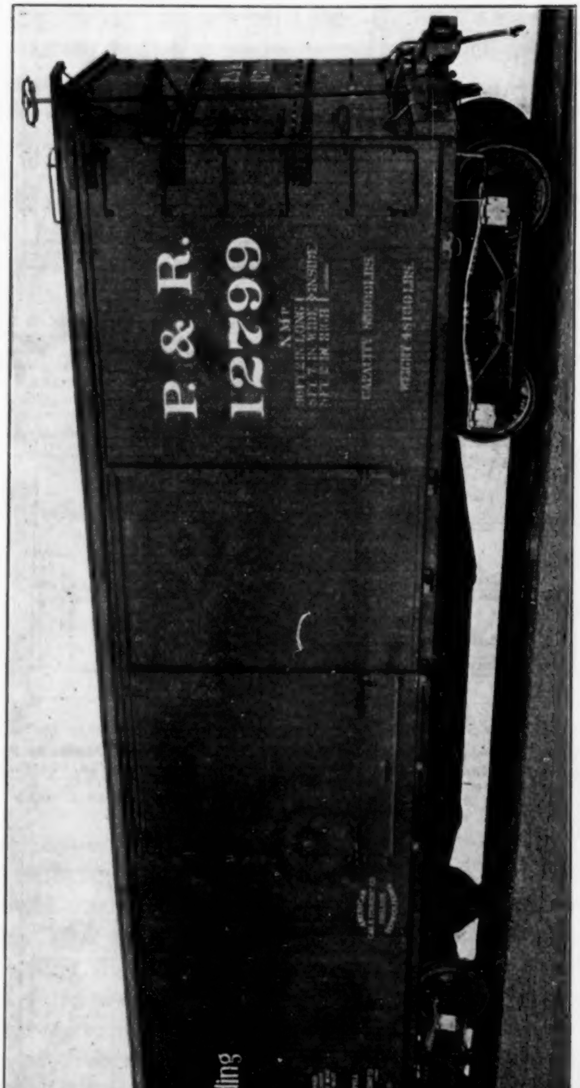
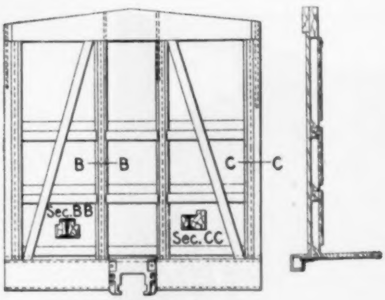
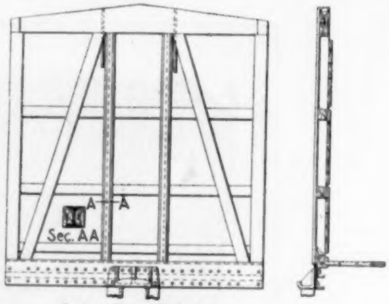



Fig. 12 40-Ton Steel Underframe Box Car used on P. & R. Ry.



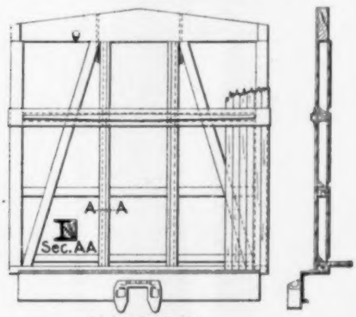

C.R.R. of N.J.

Fig. 13 End Framing used on P. & L. E. and B. & A. Cars showing I-Beam End Posts with Wood Fillers and Diagonal Braces Extending from End Sill to End Plate at Center Posts



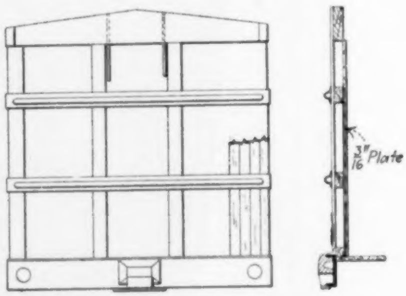

P. & L.E. B. & A.

Fig. 16 End Framing used on C. R. R. of N. J. Cars having I-Beam Center and Corner Posts with Wood Fillers; Diagonal Braces extend from End Sill to End Plate at Center Posts



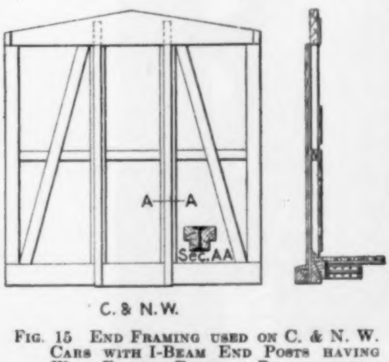

N. Y. O. & W.

Fig. 14 End Framing used on N. Y. O. & W. Cars having I-Beam End Posts with Wood Fillers; Diagonal Braces Extending from End Sill to End Plate at Center Posts and One Pressed Steel Horizontal Brace



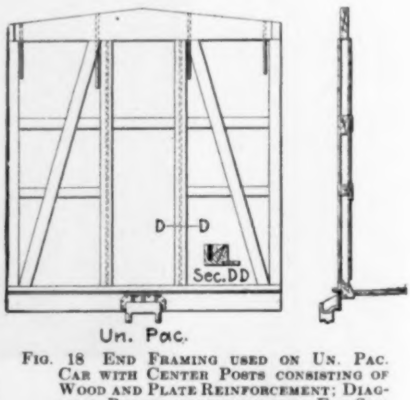

P. & R.

Fig. 17 End Framing used on P. & R. Cars with Wood Center Posts; $\frac{3}{16}$-inch End Reinforcing Plate; Two Pressed Steel Horizontal Braces




C. & N.W.

Fig. 15 End Framing used on C. & N. W. Cars with I-Beam End Posts having Wood Fillers; Diagonal Braces extend from Sill to End Plate at Center Posts




Un. Pac.

Fig. 18 End Framing used on Un. Pac. Car with Center Posts consisting of Wood and Plate Reinforcement; Diagonal Braces extending from End Sill to End Plate at Center Posts

side for securing the belt rails; this forms an ideal construction when the posts are well secured to malleable iron pockets at top and bottom, in connection with a heavy end plate and with bottom pockets riveted to the end sill cover plate. Braces have also been introduced extending, as a rule, from the end sill corner to the end plate adjacent to the posts. No doubt this location is preferable on cars with wood framing on account of the practice of extending the side brace in end panel from the bolster to the top of the corner post.

16 For securing the end frame against movement at the bottom, due to shocks occasioned by the shifting of loads, an arrangement of angle irons, located on and extending across the top of the end sill and secured by rivets to the top plate, appears to be excellent. This

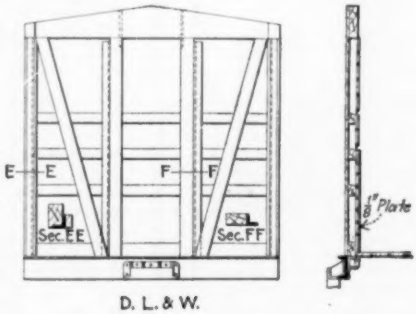



Fig. 19 End Framing used on D. L. & W. Cars showing Center and Corner Posts made up of Angle Iron and Wood Fillers; Diagonal Braces run from End Sill at Center Posts to End Plate at Corner Posts $\frac{1}{8}$-In. End Reinforcing Plate

angle iron is placed directly against the posts or sheathing. The former practice is preferable but it is necessary to cut out the sheathing at the bottom where it laps over the angle iron. Steel plates are also being introduced on the inside of box cars as shown in Fig 17, this plate extending from the floor line to the top of the upper belt rail and well secured to the corner posts by metal bands, which extend back to the door posts. It will be noted that in this construction the end posts are of wood and braced on the outside of sheathing by two pressed steel braces extending across the car, secured to the corner posts. This arrangement was recommended by the Master Car Builders' Committee on Car Construction as a desirable method for re-building the ends of wooden cars with Master Car Builders' lining.

17 It is a known fact that cars built some ten or fifteen years ago were fitted up with a type of end framing which was very light in

TABLE 6 DIMENSIONS OF END FRAMING FOR CARS WITH STEEL SIDE FRAMING

| Road | End Sills | End Plates | End Posts | End Braces | Corner Posts | Section Moduli of Posts and Braces | Type of Frame |
|---|---|---|---|---|---|---|---|
| C. & O. | 8"—11¼ lb. [ flanges out<br>¼"x14" top plate | 4"x12" wood<br>4"—5¼ lb. [ at bottom | 2—4"—7½ lb. I with<br>1—3¾"x 11/16" and<br>1—2½"x4" fillers | 2—4"—8.2 lb. Z | 4"x3"x 7/16"∠<br>5"x4" wood | 12.28 | Fig. 20 |
| Erie | 10"—15 lb. [ flanges out<br>¼"x12" top cover Plate | Pressed steel<br>Z shape | 2—4"—8.2 lb. Z | None | 5"x5"x⅜"∠ | 6.28 | Fig. 23 |
| Can. Pac. | 10"—15 lb. [ flanges out<br>¼"x12" top cover Pl. | Pressed steel<br>Z shape | 2—4"—8.2 lb. Z | None | 5"x5"x⅜"∠ | 6.28 | Fig. 23 |
| Wabash | 8"—13¾ lb. [ flanges out<br>¼"x12" top cover Pl. | Pressed steel Z<br>shape 3/16" | 2—4"—8.2 lb. Z<br>2—3"—6.7 lb. Z | None | 4"x4"x¼"∠ | 10.12 | Fig. 22 |
| Frisco | 8"—11¼ lb. [ flanges out<br>¼"x12" top cover Pl. | Pressed steel Z<br>shape | 2—4"—8.2 lb. Z<br>2—3"—6.7 lb. Z | None | 5"x4"x⅜"∠ | 10.12 | Fig. 22a |
| Un. Pac. | 9"—13¼ lb. [ flanges in<br>¼"x8" top cover Pl. | 5"x3"x 5/16"∠ | 2—4"—8.2 lb. Z | 2—4"—7¼ lb. [ | 5"x5"x⅜"∠ | 6.98 | Fig. 21 |
| P. R. R. | Pressed steel Z shape<br>upper leg 23⅝", lower leg 5⅛" | Pressed steel | 1 Pressed steel U shape | 2 Pressed steel<br>U shape | 4"x4"x⅜"∠ | 14.94 | Fig. 24 |

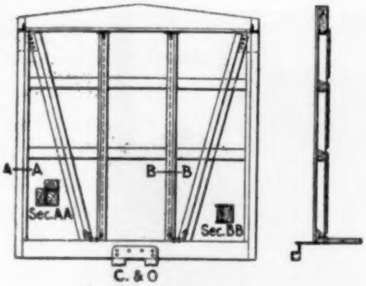



FIG. 20 STEEL END FRAMING USED ON C. & O. CARS WITH I-BEAM CENTER POSTS; DIAGONAL BRACES RUN FROM END SILL AT CENTER POSTS TO END PLATE

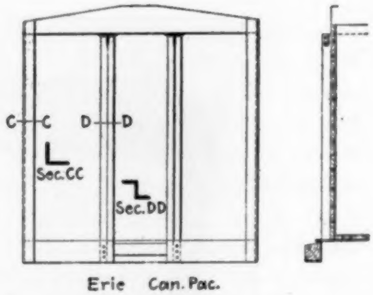



FIG. 23 STEEL END FRAMING USED ON THE ERIE AND CAN. PAC. CARS WITH Z-BAR CENTER POSTS; NO DIAGONAL BRACES.

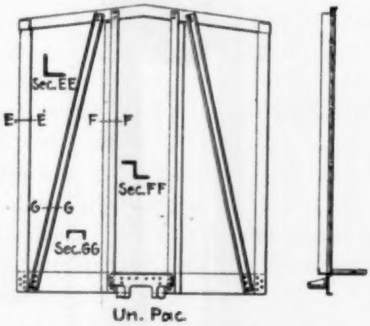



FIG. 21 STEEL END FRAMING USED ON UN. PAC. CARS WITH Z-BAR CENTER POSTS; CHANNEL BRACES RUN FROM END SILL TO END PLATE AT CENTER POSTS

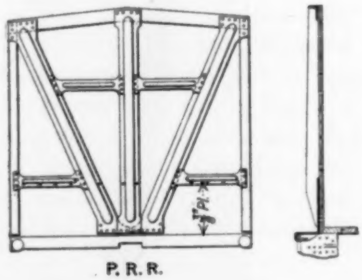
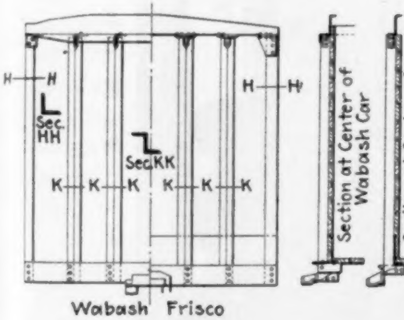



FIG. 24 STEEL END FRAMING USED ON P. R. R. CARS WITH PRESSED STEEL U-SECTIONS; DIAGONAL BRACES RUN FROM END SILL AT CENTER POSTS TO END PLATE AND ⅜-IN. END REINFORCING PLATE AT BOTTOM




FIGS. 22 AND 22a STEEL END FRAMING USED ON WABASH AND FRISCO CARS WITH Z-BAR CENTER AND INTERMEDIATE POSTS; NO DIAGONAL BRACES

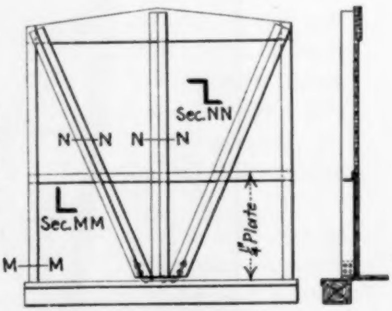



FIG 25 STEEL END FRAMING, M. C. B. RECOMMENDED DESIGN WITH SINGLE Z-BAR CENTER POST; DIAGONAL BRACES RUN FROM END SILL AT CENTER TO END PLATE AND ¼-IN. END REINFORCING PLATE AT BOTTOM

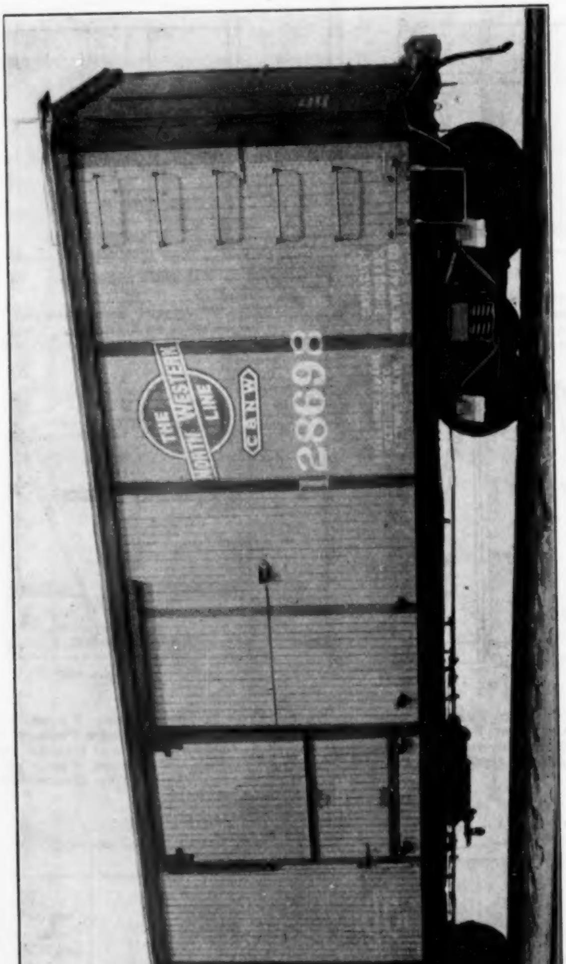



FIG. 26 40-TON STEEL UNDERFRAME BOX CAR USED ON C. & N. W. LINE

construction. No post or brace pockets were used in some cases, and in other cases where pockets were used, the metal reinforcement of the wood posts, such as channels, etc., did not extend into the pocket. When these ends were changed over to that shown in Figs. 14 and 16, with $1\frac{3}{4}$-in. end lining, no further trouble was experienced. From my personal experience, therefore, I do not believe an all-steel end is necessary to apply to wood side frame box cars.

18 *Table* 6. This shows various sizes of material used in end frame construction for cars with steel side frames, and the arrangements of posts and braces are shown in Figs. 20 to 25. Referring to Fig. 23, you will note that the diagonal brace has been omitted from the side panels; the brace has also been omitted from the end panel on the side frame and due to absence of both braces, especially the latter, I am of the opinion that the corners of the car are not sufficiently supported, and a side blow on the corner post near the eaves will result in distortion of the entire end of car. My attention was recently called to a car with end posts arranged as shown in Fig. 22; this car had received a side blow at the point in question and the entire end frame at the top took a permanent set of 4 in. from the vertical. For this reason, I believe diagonal braces extending from the side plate to the end sill near the center of car is a very desirable feature.

19 Z-bars appear to be well adapted for end posts and braces, and where set behind the end sill the best results will be obtained against strains due to end load shifting, as the entire section of metal in the Z-bar is subjected to shear. In this case it would appear desirable to use channel end sills with steel cover plate. By referring to Table 6, the variety of material used in the making of end sills may be noted. Here is where a standard construction could be made up along with striking plate, push pole pockets, corner post angles, etc., to advantage. Cars with inside sheathing should have end sills designed on standard lines, as it is a part of the car constantly requiring attention, especially in the vicinity of the end sill striking plate.

20 Referring to Fig. 25, the end construction for new cars as recommended by the Master Car Builders' Car Construction Committee may be noted. This design is evidently worked up on lines similar to those applied to the recent design of Pennsylvania box cars as shown in Fig. 24, the difference being in the use of rolled steel instead of pressed steel. I would hesitate to apply this construction to a new car, as I believe a good connection can not be made to the

TABLE 7 DIMENSIONS OF SIDE DOOR OPENINGS, SIZE OF DOOR AND DISTANCE FROM RAIL TO TOP OF FLOOR.

| Road | Distance from Rail to Top of Floor | Side Door | | | No. of Door Guide Brackets |
|---|---|---|---|---|---|
| | | Opening | Size | Type | |
| D. L. & W. | 4' 0½" | 6' 0"x7' 6½" | 6' 6"x7' 11½" | M. C. B. | 2 |
| C. R. R. of N. J. | 3' 8" | 6' 0"x7' 6½" | 6' 5½"x7' 10⅝" | M. C. B. | 4 |
| P. & L. E. | 4' 0" | 6' 0"x8' 0 1/16" | 6' 6¾"x8' 2½" | M. C. B. | 4 |
| N. Y. O. & W. | 4' 1⅛" | 6' 0"x7' 8" | 6' 5"x8' 0" | M. C. B. | 3 |
| P. & R. | 3' 7⅞" | 6' 0"x7' 6¾" | 6' 6½"x8' 2½" | Square Frame | 5 |
| B. & Albany | 4' 0" | 6' 0"x8' 0" | 6' 6¾"x8' 2½" | M. C. B. | 4 |
| C. & N. W. | 4' 1⅛" | 5' 0"x7' 5½" | 5' 2⅝"x7' 8" | Not M. C. B. | 4 |
| Un. Pac. | 3' 9 11/16" | 6' 0"x8' 8⅞" | 6' 6¾"x9' 1" | M. C. B. | 4 |
| C. & O. | 3' 11 11/16" | 5' 6"x7' 6¼" | 5' 9"x7' 9½" | Square Frame | 2 |
| Erie | 4' 1" | 6' 0 1/16"x7' 7 1/16" | 6' 3¼"x7' 8 11/16" | M. C. B. | 4 |
| Can. Pac. | 4' 2¼" | 5' 0"x7' 8" | 5' 4¾"x7' 10⅛" | M. C. B. | 2 |
| Wabash | 4' 0¼" | 6' 0"x7' 7¾" | 6' 5"x7' 10¾" | M. C. B. | 2 |
| Frisco | 4' 1" | 6' 0"x7' 7⅝" | 6' 7 1/16"x7' 9 11/16" | Not M. C. B. | 4 |
| Un. Pac. | 3' 9 11/16" | 6' 2"x8' 7" | 6' 6⅝"x8' 9 1/16" | M. C. B. | 4 |
| P. R. R. | 3' 7½" | 6' 0"x7' 3⅝" | 6' 3⅞"x7' 9¾" | ⅛" Steel | 2 |

top of the end sill of the single post and two braces as shown. Again, this framing is not adapted to the application of end doors, which a good many roads find it desirable to apply on account of the loading of lumber, rails, etc. My preference is for a similar structure, using two posts, instead of one, with a diagonal brace each side similar to Fig. 20, having all posts and braces pass below the top of the end sill, for the reasons already mentioned. The short vertical plate behind the posts is desirable, and along with the wood lining, should make a practical arrangement.

21 *Table* 7. This table gives the sizes of door openings and sizes of doors used on recently constructed box cars. Of all the things which should be made standard, a box car side door is one of the most important. The Master Car Builders' Association now has a door standard, 6 ft. 5 in. wide and 8 ft. high, and it is endeavoring to make this standard for new as well as old equipment. How many railroads are following this practice, which was recommended by their committee in 1910 and adopted as standard in 1912? By consulting this table, it will be noted that but one railroad, the New York, Ontario & Western has applied doors of this size.

22 It has been the practice to use door hangers with rollers, in order to permit the door to slide easily. Some cars were recently constructed with heavy doors, but rollers were omitted from the hangers, and I have learned from good authority that they are not satisfactory, as a great deal of exertion is required to operate them. My recommendation is to continue the use of hangers with rollers. Also sufficient number of door guides, as recommended by the Master Car Builders' Association, is not being applied; they are safeguards against accident if a door hanger should fail, and four guides should be applied as required.

23 The height from rail to top of floor shown in Table 7 varies somewhat from the Master Car Builders' standard of 4 ft. There is no reason for this oversight on the part of motive power officials, and it is of considerable importance as it permits of standard unloading platform heights all over the country, making it easier for electric freight handling trucks to enter the cars on the same level.

24 *Table* 8. This table gives some idea of the variety of roofs and roof carlines applied to recent box cars. The number of carlines required seems to be a disputed matter and evidently has been left to the manufacturer who supplies the roof. The essential requirements for a carline are to support the weight of the roof, with a suitable allowance for additional loading due to accumulation of snow

TABLE 8 DIMENSIONS OF RUNNING BOARDS, TYPE OF ROOF AND CARLINES, WIDTH AND HEIGHT AT EAVES, ETC.

| Road | Type of Carline | No. of Carlines | Type of Roof | Max. Width over Eaves | Distance, Rail to Eaves | Running Board | | |
|---|---|---|---|---|---|---|---|---|
| | | | | | | Width | Size of Matl. | Length |
| D. L. & W. | Pressed Steel ∪; also ∩ | 7 | Inside and all Steel | 9′ 7″ | 12′ 5½″ | 19″ | 1⅛″x6″ | 38′ 2¼″ |
| C. R. R. of N. J. | Pressed Steel | 7 | Inside Metal | 9′ 6¾″ | 12′ 2⅞″ | 19″ | 1″x6″ | 38′ 6″ |
| P. & L. E. | Rolled Steel | 11 | All Steel | 9′ 5½″ | 12′ 6½″ | 20″ | 1″x6″ | 38′ 3¼″ |
| N. Y. O. & W. | Pressed Steel [ | 11 | All Steel | 9′ 6″ | 12′ 5½″ | 24″ | 1″x7½″ | 38′ 8″ |
| P. & R. | Pressed Steel ∪ | 8 | Inside Metal | 9′ 10″ | 12′ 1¾″ | 19″ | 11/16″x5¾″ | 38′ 7″ |
| B. & Albany | Pressed Steel [ | 8 | Outside Metal | 9′ 5 1/16″ | 12′ 7⅛″ | 20″ | 1″x6″ | 38′ 3″ |
| C. & N. W. | ⅜″x3″ Plate; 3″x3″ wood | 12 | Inside Metal | 9′ 9″ | 12′ 5⅞″ | 18½″ | 1¼″x6″ | 41′ 10″ |
| Un. Pac. | 4″x2″x¼″ ∠; 1⅞″x9¾″ wood | 9 | Outside Metal | 10′ 1″ | 13′ 4 3/16″ | 24″ | 1⅛″x7½″ | 42′ 5″ |
| C. & O. | 3″x3″x¼″ ∠ and wood | 7 | Outside Metal | 9′ 7½″ | 12′ 4″ | 24″ | 1″x7½″ | 38′ 3⅝″ |
| Erie | Pressed Steel [ | 8 | Inside Metal | 9′ 3⅛″ | 12′ 6 11/16″ | 19″ | 1¼″x6″ | 38′ 5½″ |
| Can. Pac. | Pressed Steel [ | 8 | Inside Metal | 9′ 3½″ | 12′ 7 3/16″ | 24″ | ⅞″x7½″ | 38′ 5½″ |
| Wabash | Pressed Steel ∩ | 11 | All Steel | 9′ 10½″ | 12′ 0½″ | 18¾″ | ⅞″x5¾″ | 38′ 7″ |
| Frisco | 4″—9½ lb. I | 8 | Outside Metal | 9′ 7⅛″ | 12′ 5⅝″ | 18″ | 1⅛″x5¾″ | 42′ 1″ |
| Un. Pac. | 3″x3″x¼″ ∠, and wood | 9 | Outside Metal | 9′ 8″ | 13′ 1 3/16″ | 24″ | 1⅛″x7½″ | 42′ 5″ |
| P. R. R. | Pressed Steel ∪ | 10 | ⅛″ Steel Plate, riveted | 9′ 11 11/16″ | 11′ 8″ | 18⅛″ | 1½″x5¾″ | 42′ 11½″ |

and ice, and which is of equal importance, to prevent the sides of the car near the eaves from spreading due to lading.

25 As car roofs are now receiving considerable attention by motive power officials, I will describe the various types in use. They are usually divided into three classes, known as inside roofs, outside roofs, and all-steel roofs. The classification, inside roof, covers all types of car roofs where the metal is inside of the car, and the roof boards over this metal are on the outside of the car. The classification, outside metal roof, covers car roofs where the steel is on the outside of the car, and the roofing boards are underneath. The classification, all-steel roof, covers a car roof in which a heavy sheet is used, usually about 16 U. S. gage material. In this case, no roof boards are used, either above or under the roofing sheets.

26 The inside metal roof is the oldest type of roof in service. The original freight car roof consisted of either single or double sheathing of boards on top of the car, but the strain on these roofs was such that they could not be kept without continual repairs. The inside metal roof was then developed and being made of separate flexible units, permitted weaving of the car to take place and still have a waterproof roof below the roof boards. The outside roof boards are simply a protection to keep the trainmen from walking on the roofing sheets. The advantages of the inside type are: (*a*) flexibility, (*b*) protection of the metal from the weather, (*c*) protection of the metal from trainmen's boots, and (*d*) prevention of the quick corrosion of the metal, due to the roof boards preventing a deposit of cinders on the metal, which, when attacked by moisture, quickly eats away the galvanizing and also the sheet itself.

27 The objections raised against this type are that the outside roofing boards work loose, the torsional strains of the car causing the nails to work out and rust off. This condition is usually caused by the weaving of the car superstructure and can be eliminated to some extent by the use of stronger carlines. An inside roof properly applied as far as the metal is concerned, should last a good many years, with the replacement of the roof boards about every four years. These roof boards, besides being secured by nails, are now held down at the eaves by wire staples locking the boards to the facia.

28 Of the outside metal roofs, there are numerous types on the market, but in a general way they are similar. Their differences usually are in the method of fastening the sheets at the eaves. The gage of metal ranges in most cases from 22 to 26 U. S. The advantages claimed for the outside metal roofing are: (*a*) ease of ap-

TABLE 9 DISTANCE BETWEEN BODY BOLSTERS, SPREAD OF TRUCK WHEELS, TYPE OF TRUCK FRAMES, BOLSTERS, ETC.

| Road | Inside Length | Length over Striking Plate | Truck | | | | |
|---|---|---|---|---|---|---|---|
| | | | Centers | Wheel Base | Type of Frame | Type of Bolster | Kind of Wheels |
| D. L. & W. | 36′ 0″ | 37′ 7⅝″ | 27′ 0″ | 5′ 0″ | Arch Bar | Cast Steel | Cast Iron |
| C. R. R. of N. J. | 36′ 0″ | 38′ 3½″ | 27′ 0″ | 5′ 6″ | Arch Bar | Cast Steel | Cast Iron |
| P. & L. E. | 36′ 0″ | 38′ 0½″ | 27′ 0″ | 5′ 6″ | Cast Steel | Steel Built up | Cast Iron |
| N. Y. O. & W. | 36′ 0″ | 38′ 8⅞″ | 26′ 6″ | 5′ 6″ | Cast Steel | Cast Steel | Cast Iron |
| P. & R. | 36′ 2″ | 38′ 2½″ | 27′ 6″ | 5′ 6″ | Arch Bar | Steel Built up | Cast Iron |
| B. & Albany | 36′ 0″ | 38′ 1″ | 27′ 0″ | 5′ 6″ | Cast Steel | Steel Built up | Cast Iron |
| C. & N. W. | 40′ 0″ | 41′ 10⅞″ | 30′ 6″ | 5′ 0″ | Arch Bar | Steel Built up | Cast Iron |
| Un. Pac. | 39′ 11⅞″ | 42′ 1¼″ | 30′ 8″ | 5′ 6″ | Cast Steel | Steel Built up | Cast Iron |
| C. & O. | 36′ 0″ | 38′ 6½″ | 26′ 10″ | 5′ 6″ | Arch Bar | Cast Steel | Cast Iron |
| Erie | 36′ 0″ | 38′ 1¼″ | 26′ 10″ | 5′ 4″ | Cast Steel | Steel Built up | Cast Iron |
| Can. Pac. | 36′ 0″ | 38′ 1¼″ | 26′ 10″ | 5′ 4″ | Arch Bar | Steel Built up | Cast Iron |
| Wabash | 36′ 5⅝″ | 38′ 5⅝″ | 27′ 2″ | 5′ 4″ | Arch Bar | Steel Built up | Cast Iron |
| Frisco | 40′ 0″ | 42′ 0¾″ | 31′ 0″ | 5′ 6″ | Cast Steel | Cast Steel | Cast Iron |
| Un. Pac. | 40′ 5⅜″ | 42′ 1¼″ | 30′ 8″ | 5′ 6″ | Cast Steel | Steel Built up | Cast Iron |
| P. R. R. | 40′ 6″ | 42′ 6″ | 32′ 6″ | 5′ 6″ | Arch Bar | Cast Steel | Rolled Steel |

plication, (*b*) ease of repair, (*c*) somewhat lower first cost, and (*d*) possible prevention of fires, particularly in the west.

29 The objections which develop in this type of roof have been: the exposure of the metal itself to weather and chafing of the sheets where joined, causing the galvanizing to wear off so that the sheet quickly rusts. Cinders also collect along the eaves where castings are used, and when wet, sulphuric acid action will set in and eat out the sheet. This is partially overcome by the use of a slip joint along the eaves as provided for on the later types.

30 The all-steel type of roof is a development of the last three or four years, and consists of metal plates of about 16 U. S. gage resting on steel carlines, they being an integral part of the roof. In some cases the sheets extend from side to side of the car, and in others the sheets are divided extending from the eaves to the ridge pole. The amount of galvanizing used on these sheets is practically the same as that on the outside metal roofs, which is about 1.8 oz. per sq. ft. on both sides of the sheet. The claim is made that this type somewhat lightens the weight of the car, but the same objections have been raised regarding the wearing out of the material as well as the tearing of the metal due to its rigid application. This latter feature is now receiving some attention by the manufacturers, and no doubt can be overcome. This style of roof is more expensive than the other types previously mentioned.

31 Large sums of money are paid out annually by our railroads to cover damage claims due to wet lading and the roof problem is a severe one with us. The service requirements, due to the use of heavy power and the severe service that cars are receiving in freight classification yards, compel us to give this matter our earnest attention and assist the manufacturers of car roofs as much as possible in developing a roof that, with ordinary care, will stay tight and protect the lading. Considerable success has been obtained with roofs applied to box cars made up of inside and outside roof boards, having a layer of waterproof roofing material between the boards, similar to that applied to refrigerator cars.

32 In the application of running boards, there seems to be no standard method followed. Table 8 shows that the width over all varies from 18 in. to 24 in. and size of material from 7/8 in. to 1½ in. thick and in widths from 5¾ in. to 7½ in. This does not appear on the surface to be a very important matter, but it seems to me that a better understanding should have existed between railroads, at least to arrive at a standard width of platform for the safety of train-

TABLE 10 DIMENSIONS OF DRAFT SILLS, SIDE SILLS, FLOOR SUPPORTS, ETC.

| Road | Side Sills | Draft Sills | End Sill Diagonal Brace | Body Bolster | | | Cross Bearer | | | | Floor Beams | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Type | Plates Top | Plates Bottom | Type | Plates Top | Plates Bottom | Spacing Ctr. to Ctr. | Type | No. |
| D. L. & W. | 8"—11¼ lb[ with 3"x3½"x5/16" ∠ 3¾"x4⅜" wood | 12"—25 lb. [ | 4"—8.2 lb. Z | Pressed Steel ][ | ½"x20½" | ⅜"x10" | Cast Steel | 5/16"x7" | 5/16"x6" | 7' 0" | 4"—8.2 lb. Z | 3 |
| C.R.R. of N.J. | 12"—20½ lb. [ 3"x3½"x⅜"∠ 3¾"x4" wood | 15"—33 lb. [ | 5"—6.5 lb. [ | ¼" Pressed Steel [ s back to back | 7/16"x17½" | 7/16"x14" | ¼" Pressed Steel [ | ⅜"x6" | ⅜"x6" | 9' 0" | 6"—8 lb. [ | 3 |
| N.Y.O.& W. | ¼" Pressed Steel Z—9" high | ⅜" Pressed Steel Z | 5"—6.5 lb. [ | ¼" Pressed Steel ][ | ⅜"x14" | ½"x14" | ¼" Pressed Steel [ | ¼"x6" | ¼"x6" | 8' 7" | ¼" Pressed Steel [ | 3 |
| P. & R. | 10"—20 lb. [ 3"x3"x5/16"∠ 3½"x4" wood | ⅜" Pressed Steel Z spliced to ctr. sill | 5"—9 lb. [ | ⅜" Pressed Steel [ s back to back | ½"x24" | ¾"x16" | ⅜" Pressed Steel [ | ½"x6" | ⅝"x6" | 9' 2" | 5"—9 lb. [ | 3 |
| B & Albany | 4"—8.2 lb. Z 4"x4" wood | ⅜" Pressed Steel Z spliced to ctr. sill | 5"x3"x5/16" ∠ | ¼" Pressed Steel ][ | ⅜"x15" | ⅜"x15" | ¼" Pressed Steel [ | ¼"x3½" | ¼"x3¾" | 9' 0" | Same as cross-bearer with cover Plates | 3 |
| C. & N. W. | 6"—15.6 lb. Z | 15"—33 lb. [ | 3½"x3½"x⅜" ∠ | ¼" Pressed Steel ][ | ⅜"x22½" | ⅝"x15" | ¼" Pressed Steel [ | ⅜"x5½" | ⅜"x5½" | 7' 8" | 4"—9½ lb. I | 8 |

TABLE 10—Continued

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Un. Pac. | 5"—11.6 lb. Z with 3 11/16"x5½" wood | ⅜" Pressed Steel spliced to ctr. sill | None | Cast Steel | ........ | ........ | 10"—25 lb I | ........ | ........ | 9' 0" | 10"—25 lb. I | 3 |
| C. & O. | 8"—11¼ lb. [ 5½"x3" wood | 15"—33 lb. [ | ¼" Pressed Steel [ 5" wide | ¼" Pressed Steel ][ | ½"x12" | ⅝"x12" | ¼" Pressed Steel [ | ⅜"x6" | ⅜"x6" | 6' 6" | 4"—5¼ lb. [ | 3 |
| Erie | 8"—11¼ lb. [ flanges in | 15"—33 lb. [ | 5"x3½"x⅜" ∠ | ¼" Pressed Steel ][ | ½"x15" | ½"x15" | ¼" Pressed Steel [ | ½"x6" | ½"x6" | 5' 3½" | 4"—8.2 lb. Z | 4 |
| Can. Pac. | 8"—11¼ lb. [ flanges in | 15"—33 lb. [ | 5"x3½"x⅜" ∠ | ¼" Pressed Steel ][ | ½"x15" | ½"x15" | ¼" Pressed Steel [ | ½"x6" | ½"x6" | 5' 5½" | 4"—8.2 lb. Z | 4 |
| Wabash | 8"—11¼ lb. [ flanges in | ⅜" Pressed Steel Z spliced to ctr. sill | 4"x3"x¼" ∠ | ¼" Pressed Steel ][ | ⅜"x15" | ⅜"x15" | ¼" Pressed Steel [ | ⅜"x6" | ⅜"x6" | 9' 0" | ¼" Pressed Steel [ | 3 |
| Frisco | 5"x3"x⅜" ∠ 3"x4⅝" wood | ⅜" Pressed Steel Z spliced to ctr. sills | 5"—6½ lb. [ | 5/16" Pressed Steel ][ | ⅜"x14½" | ⅜"x18" | 9"—15 lb. [ | ........ | ........ | 6' 5¾" | 9"—15 lb. [ | 2 |
| Un. Pac. | 9"—13¼ lb. [ flanges in | ¼" Pressed Steel outside angle No splice | None | ¼" Pressed Steel ][ | ⅜"x14" | ⅜"x14" | ¼" Pressed Steel [ | ¼"x6" | ¼"x6" | 9' 0" | ¼" Pressed Steel [ | 3 |
| P. R. R. | 6"x4"x⅜" ∠ | ⅜" Pressed Steel No splice | Pressed Steel [ 8" wide | ⅜" Pressed Steel U Shape | ........ | ........ | ⅜" Pressed Steel ][ | ⅜"x12" | ⅜"x12" | 17' 10" | Pressed Steel [ ¾" deep | 6 |

men and standard size of materials. The Interstate Commerce Commission recommends a standard width of 20 in. and allows a minimum of 18 in. I would suggest, therefore that all railroads, when applying new running boards to old or new equipment, construct the platform 20 in. wide using $1\frac{1}{8}$ in. by $6\frac{1}{4}$ in. material.

33 *Table* 9. This table shows the spread of body bolsters for box cars of different inside lengths, also distance over striking plates, truck wheel centers, etc. For cars 36 ft. inside length, bolster centers vary from 26 ft. 6 in. to 27 ft. 6 in. and for the longer car this variation is shown from 30 ft. 6 in. to 32 ft. 6 in. The effect of a long overhang of the car from the center line of the bolster to the pulling face of the coupler has a tendency to cause derailment on curves, due to the coupler side clearance in the striking plate not being sufficient to permit the coupler to swing over when the car is coupled with another car that has a short overhang; if derailment does not occur, excessive wheel flange wear is produced. For 36-ft. box cars it would appear desirable to have the bolster centers made 27 ft. which is now standard on a large number of railroads and allows ample space for the application of draw gear.

34 For longer cars the same overhang as obtained under the above conditions for 36-ft. cars could be maintained, which would make the bolster centers 31 ft. for a car 40 ft. long inside. The distance from the face of the striking plate to the outside face of the body end construction should be ample to permit of the proper clearances being maintained for safety appliances, as required by law.

35 Truck wheel centers vary somewhat, the majority of the railroads now using trucks with wheels spaced 5 ft. 6 in. apart. This I consider very good practice, as it allows ample room for inside brake rigging. Truck arch bars and cast-steel side frames should be designed to interchange with one another and should conform to the Master Car Builders' requirements. The practice of using various designs of cast-steel side frames which vary in details should be discouraged, as there is no reason why a standard design of truck side frame for the different capacity cars should not be obtained and applied by every railroad in the country. This certainly would simplify repairs and they could then be purchased in the open market subject to specifications.

36 Bolster designs should also be made standard. This is a very important matter as the life of truck bolsters made of cast steel is uncertain, a number being replaced within five to ten years after cars

TABLE 11 CARS WITH WOOD SIDE FRAMES AT CENTER

| Road | Sill Type | Section Type | Depth over Flanges, In. | Web Plate, In. | Outside Top Angle, In. | Bottom Angle, In. | | Cover Plate, In. | | Min. Area, Sq. In. | Neutral Axes | | Section Moduli | | Max. Stress due to Vertical Loads, Lb. per Sq. In. | Compression due to End Shock, Lb. per Sq. In. | Compression due to Eccentricity, Lb. per Sq. In | Maximum Combined Compression Lb. per Sq. In. | Center Line of Draw Gear above or below Neutral Axis, In. | $\frac{1}{A}+\frac{e}{S}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Outside | Inside | Top | Bottom | | $X_c$ | $X_t$ | $S_c$ | $S_t$ | | | | | | |
| D. L. & W. | A | A | 12 | .... | ........ | ........ | ........ | 2–¼x19 | 2–¼x19 | 32.12 | 6.21 | 6.79 | 154.16 | 141.0 | 9851c<br>10770t | 9338 | 6402 | 12787 | 3.29<br>below | 0.054 |
| C. R. R. of N. J. | A | A | 15 | .... | ........ | ........ | ........ | 1–5/16x20½<br>1–¼x20½ | 1–5/16x20½<br>1–¼x20½ | 36.23 | 7.67 | 8.20 | 209.55 | 196.0 | 7421c<br>7885t | 8279 | 697 | 16401 | .487<br>above | 0.029 |
| N. Y. O. & W. | B | G | 24 | ¼ | 3⅞fl'ge | 3½x3x 7/16 | 3½x3x 7/16 | ¼x21¼ | ........ | 27.67 | 12.74 | 11.51 | 212.57 | 235.28 | 8462c<br>7645t | 10842 | 2455 | 21759 | 1.74<br>above | 0.0442 |
| P. & R. | C | E | 26 | ⅜ | 4x4x½ | 4x4x½ | 4x4x½ | ¼x22½ | ........ | 45.01 | 13.06 | 13.19 | 377.69 | 373.97 | 5514c<br>5569t | 6664 | 5012 | 17190 | 6.31<br>above | 0.038 |
| B. & A. | D | E | 22 | ¼ | 3x3x⅜ | 3x3x⅜ | 3x3x⅜ | ¼x20¼ | ........ | 25.84 | 10.18 | 12.07 | 213.67 | 180.21 | 8931c<br>10590t | 11177 | 3410 | 23519 | 2.43<br>above | 0.0485 |
| C. & N. W. | E | B | 15 | .... | ........ | ........ | ........ | ⅜x20 | ⅜x20 | 33.12 | 7.51 | 8.24 | 188.57 | 171.86 | 11859c<br>13012t | 9056 | 7143 | 13772 | 4.49<br>below | 0.058 |
| Un. Pac. | F | E | 30½ | ¼ | 3x3x 5/16 | 3x3x⅜ | 3x3x 5/16 | ¼x20½ | ........ | 32.09 | 15.26 | 15.49 | 317.92 | 313.2 | 8566c<br>8694t | 9346 | 5671 | 23583 | 6.01<br>above | 0.049 |

are placed in service. It would avoid unnecessary delay in obtaining new bolsters for repairs when replacing a built-up design, especially when changing from one manufacturer to another, who not having a bolster pattern that will interchange, will try to furnish something else or make new patterns and in the meanwhile cars are tied up at the repair shops. It is manifestly impossible to carry such parts in stock for five to ten different designs for the same capacity car, as this cannot be done without increasing the amount of stock 25 to 50 per cent over the present allowance.

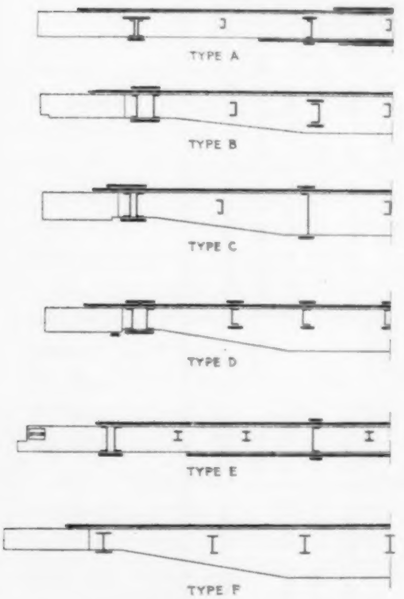



Fig. 27

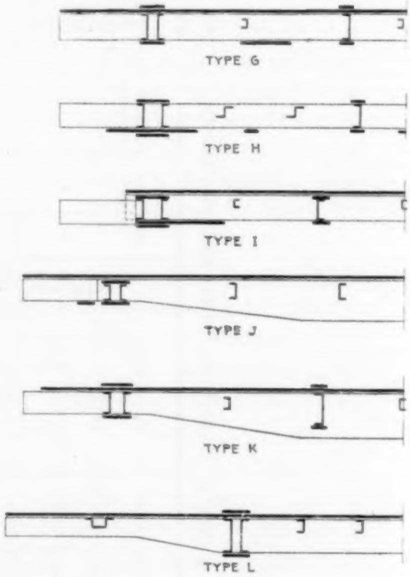



Fig. 28

Fig. 27 Types of Center Sills showing the Diversity in Design, Arrangement of Cross Bearers, Length of Cover Plates, etc., as applied to Box Cars with Wood Side Frames

Fig. 28 Types of Center Sills showing the Diversity in Design, Arrangement of Cross Bearers, Length of Cover Plates, etc., as applied to Box Cars with Steel Side Frames

37 *Tables* 11 to 14. These tables have been compiled to show interesting information on the subject of center sill constructions for 14 different cars, the types of underframes used in which are shown in outline on Figs. 27, 28 and 31. The heavy lines in the drawings indicate the cover plates used which appear to be a general practice among

TABLE 12 CARS WITH STEEL SIDE FRAMES AT CENTER

| Road | Sill Type | Section Type | Depth over Flanges, In. | Web Plate, In. | Outside Top Angle, In. | Bottom Angle, In. | | Cover Plate, In. | | Min. Area, In. Sq. In. | Neutral Axes | | Section Moduli | | Max. Stress due to Vertical Loads Lb. per Sq. In. | Compression due to End Shock, Lb. per Sq. In. | Compression due to Eccentricity, Lb. per Sq. In. | Maximum Combined Compression Lb. per Sq. In. | Center Line of Draw Gear above or below Neutral Axis, In. | $\frac{1}{A}+\frac{e}{S}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Outside | Inside | Top | Bottom | | $X_c$ | $X_t$ | $S_c$ | $S_t$ | | | | | | |
| C. & O. | G | C | 15 | .... | ........ | ........ | ........ | $\frac{5}{16}$x19¾ | ........ | 25.96 | 6.00 | 9.31 | 150.03 | 96.69 | 10972 c<br>17025 t | 7704 | 7345 | 11331 | 5.56 below | 0.095 |
| Erie | H | D | 15 | .... | ........ | ........ | ........ | ........ | ........ | 18.73 | 7.092 | 7.908 | 80.0 | 71.72 | 20311 c<br>22655 t | 10678 | 10275 | 20714 | 4.158 below | 0.1112 |
| Can. Pac. | H | D | 15 | .... | ........ | ........ | ........ | ........ | ........ | 18.73 | 7.092 | 7.908 | 80.0 | 71.72 | 20311 c<br>22655 t | 10678 | 13270 | 17719 | 5.408 below | 0.1287 |
| Wabash | I | C | 15 | .... | ........ | ........ | ........ | $\frac{5}{16}$x20 | ........ | 26.05 | 5.98 | 9.33 | 151.04 | 96.8 | 10887 c<br>16987 t | 7677 | 6806 | 11758 | 5.14 below | 0.091 |
| Frisco | J | E | 26 | ¼ | 3x3x¼ | 3x3x⅜ | 3x3x$\frac{5}{16}$ | ¼x20 | ........ | 26.89 | 12.38 | 13.87 | 235.53 | 210.23 | 8725 c<br>9775 t | 7437 | 747 | 16909 | .88 above | 0.04 |
| U. P. | K | G | 27¾ | ¼ | 3⅞ fl. | 3½x3x⅜ | 3½x3x⅜ | ¼x23 | ........ | 28.78 | 13.94 | 14.06 | 261.67 | 259.43 | 9053 c<br>9131 t | 6949 | 3829 | 19831 | 5.0 above | 0.053 |
| P. R. R. | L | H | 20 | ⅜ | 4 fl. | 4 fl. | 4x4x$\frac{5}{16}$ | ⅜x24 | ........ | 35.91 | 9.55 | 10.82 | 258.66 | 228.30 | 10600 c<br>12013 t | 5569 | 2265 | 18434 | 2.93 above | 0.038 |

designers. This is considered by many to be a very desirable feature and besides adding to the rigidity of the structure, increases the net area of section and therefore reduces the fiber stress to within reasonable figures.

38 The strains which an underframe has to withstand vary, and depend a great deal on the design and arrangement of body bolsters, side sills and body framing. With substantial side sills and body bolsters, compressive strains can be transmitted by the latter and thus part of the compressive load can be taken care of by the side sills. However, there is a limit to the load the side sills can take care of which is based on the ratio of length to the least radius of gyration, the length being considered as the maximum distance between adjacent floor beams. In the case of cars with wood body side framing it would appear desirable to provide sufficient area at the smallest section of the center sills to take care of end strains, allowing the side sill to carry only its proportions of the vertical loads due to lading, weight of superstructure and an allowance for oscillation.

39 In the case of box cars with steel side frames there is no question about the ability of the frame structure to carry considerably greater loads, both vertically and due to end shock, on account of the diagonal braces assisting in transmitting the strain throughout the side. For this reason some designers do not consider it necessary to use continuous cover plates on the center sills, but use rolled steel sections for these sills, having a much smaller net area than what is considered good practice by others.

40 In order to analyze properly the stresses in underframes a standard method of comparison should be made. Cars do not fail as a rule because of the weight of lading, but principally because of strains transmitted through the coupler. The magnitude of the end shocks that underframes have to withstand were investigated in tests conducted prior to 1902, on the Lake Shore & Michigan Southern Railway with a dynamometer car having nominal capacity in apparatus of 300,000 lb. It was found that the tensile and buffing strains, with an engine having a tractive power of 36,000 lb., were from 50,000 to 70,000 lb. and 80,000 to 150,000 lb. respectively, depending upon the skill of the engineer in manipulating the engine, the train remaining intact. In coupling an engine to its train, buffing strains from 65,000 to 142,000 lb. were obtained. Thirty cars moving at about 6½ miles per hour, and coupling on to ten loaded cars with brakes set, gave a shock of 376,492 lb. It would therefore appear from the above results, that provisions should be made in designing a steel

TABLE 13 CARS WITH WOOD SIDE FRAMES AT BOLSTER

| Road | Sill Type | Section Type | Depth over Flanges, In. | Web Plate, In. | Outside Top Angle, In. | Bottom Angle, In. | | Cover Plate, In. | | Min Area, Sq. In. | Neutral Axes | | Section Moduli | | Compression due to End Shock, Lb. per Sq. In. | Compression due to Eccentricity, Lb. per Sq. In. | Maximum Combined Compression Lb. per Sq. In. | Center Line of Draw Gear above or below Neutral Axis | $\frac{1}{A}+\frac{e}{S}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Outside | Inside | Top | Bottom | | $X_c$ | $X_t$ | $S_c$ | $S_t$ | | | | | |
| D. L. & W. | A | B | 12 | ... | ... | ... | ... | ¼x19 | ¼x19 | 24.20 | 6.25 | 6.25 | 103.11 | 103.11 | 12396 | 8728 | 21124 | 3.00 below | 0.07 |
| C. R. R. of N. J. | A | C | 15 | ... | ... | ... | ... | ¼x20½ | ... | 23.91 | 6.33 | 8.92 | 134.27 | 95.28 | 12547 | 2109 | 14656 | 0.67 below | 0.048 |
| N. Y. O. & W. | B | G | 15 | ¼ | 3⅞ fl'ge | 3½x3x1/16 | 3½x3x1/16 | ¼x21¼ | ... | 25.00 | 8.42 | 6.83 | 119.84 | 147.76 | 12000 | 5238 | 17238 | 2.58 below | 0.057 |
| P. & R. | C | E | 11¼ | ⅜ | 4x4x½ | 4x4x½ | 4x4x½ | ¼x22½ | ... | 36.18 | 5.91 | 5.59 | 123.53 | 130.60 | 8291 | 1929 | 10220 | 0.84 below | 0.033 |
| B. & A. | D | E | 12⅞ | ¼ | 3x3x⅜ | 3x3x⅜ | 3x3x⅜ | ¼x20¼ | ... | 23.90 | 6.28 | 6.84 | 110.31 | 101.13 | 12552 | 4331 | 16883 | 1.46 below | 0.055 |
| C. & N. W. | E | C | 15 | ... | ... | ... | ... | ⅜x20 | ... | 27.30 | 5.76 | 9.61 | 164.24 | 98.44 | 10989 | 18986 | 29975 | 6.23 below | 0.099 |
| Un. Pac. | F | F | 12⅞ | ¼ | 3x3x5/16 | 3x3x⅜ | ... | ¼x20½ | ... | 19.08 | 5.11 | 8.01 | 97.61 | 62.33 | 15719 | 19901 | 35620 | 4.135 below | 0.118 |

underframe box car to take care of an impact blow of 350,000 lb. transmitted throughout all sills.

41 It would therefore seem advisable to assume an end strain of 200,000 lb. on the center sills of box cars with steel side frames and 300,000 lb. for box cars with wood side frames for the reasons previously mentioned. If the center line of coupler was directly on line with the neutral axis of the section to be analyzed, the stress per square inch on the center sills due to end shocks would be equal to the buffing force $B$ divided by the area $A$ of section. Referring to the data compiled in Tables 11 to 14 on center sills, the area of the sills are given both at the center line of the car and near the bolster. The eccentricity $e$ or the distance from the center line of the draw gear to the neutral axis is also given for these sections. Stress at the bolster due to eccentricity equals $\frac{B_e}{S_t}$ when the center line of the draw gear is below the neutral axis, and $\frac{B_e}{S_c}$ when the center line is above the neutral axis. The combined stress at the bolster equals $\frac{B}{A} + \frac{B_e}{S_t}$ or $\frac{B}{A} + \frac{B_e}{S_c}$, depending upon the location of the center line of the draw gear. At the center line of the car, compression stress due to the lading equals $\frac{M}{S_c}$. Stress due to eccentricity equals $\frac{B_e}{S_c}$. Stress due to end shock equals $\frac{B}{A}$. When the center line of the draw gear at the center of the car is below the neutral axis, the combined stress equals $\frac{M}{S_c} + \frac{B}{A} - \frac{B_e}{S_c}$, and if the center line is above, the combined stress equals $\frac{M}{S_c} + \frac{B}{A} + \frac{B_e}{S_c}$. The ratio of stress to end strain is obtained by the formula $\frac{1}{A} + \frac{e}{S}$, in which $S$ represents the section modulus, $S_t$ to be used if the center line of the draw gear is below, and $S_c$ if above, the neutral axis. The Car Construction Committee of the Master Car Builders' Association recommend that the above ratio on new cars should not exceed 0.06. They also recommend a minimum area of 24 sq. in. of center sills.

42 Tables 11 and 12 give the maximum stress on center sills due to vertical loading which was obtained by assuming a uniform load distributed throughout the sill. Calculations are based on the weight of car body, $A$, the lading, $B$, and the oscillation, $C$. The car body,

TABLE 14 CARS WITH STEEL SIDE FRAMES AT BOLSTER

| Road | Sill Type | Section Type | Depth over Flanges, In. | Web Plate, In. | Outside Top Angle, In. | Bottom Angle, In. | | Cover Plate, In. | | Min. Area, Sq. In. | Neutral Axes | | Section Moduli | | Compression due to End Shock, Lb. per Sq. In. | Compression due to Eccentricity, Lb. per Sq. In. | Maximum Combined Compression Lb. per Sq. In. | Center Line of Draw Gear above or below Neutral Axis, In. | $\frac{1}{A}+\frac{e}{S}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Outside | Inside | Top | Bottom | | $X_c$ | $X_t$ | $S_c$ | $S_t$ | | | | | |
| C. & O. | G | C | 15 | ..... | ..... | ..... | ..... | 5/16x19¾ | ..... | 25.96 | 6.00 | 9.31 | 150.03 | 96.69 | 7704 | 11500 | 19204 | 5.56 below | 0.095 |
| Erie | H | D | 15 | ..... | ..... | ..... | ..... | ..... | ..... | 19.8 | 7.5 | 7.5 | 83.4 | 83.4 | 10101 | 8993 | 19094 | 3.75 below | 0.095 |
| Can. Pac. | H | D | 15 | ..... | ..... | ..... | ..... | ..... | ..... | 19.8 | 7.5 | 7.5 | 83.4 | 83.4 | 10101 | 11990 | 22091 | 5.0 below | 0.110 |
| Wabash | I | C | 15 | ..... | ..... | ..... | ..... | 5/16x20 | ..... | 26.05 | 5.98 | 9.33 | 151.04 | 96.8 | 7677 | 10620 | 18297 | 5.14 below | 0.091 |
| Frisco | J | E | 13⅞ | ¼ | 3x3x¼ | 3x3x⅜ | 3x3x5/16 | ¼x20 | ..... | 22.34 | 6.93 | 7.19 | 108.01 | 104.1 | 8953 | 8760 | 17713 | 4.56 below | 0.088 |
| Un. Pac. | K | G | 13 | ¼ | 3⅞ fl. | 3½x3x⅜ | 3½x3x⅜ | ¼x23 | ..... | 23.03 | 6.86 | 6.39 | 103.54 | 111.16 | 8684 | 4174 | 12858 | 2.32 below | 0.063 |
| P. R. R. | L | H | 11 | ⅜ | 4 fl. | 4 fl. | 4x4x9/16 | ⅜x24 | ..... | 30.68 | 5.38 | 5.99 | 125.62 | 112.83 | 6517 | 2198 | 8715 | 1.24 below | 0.043 |

weight $A$, carried on center sills was taken at 20,000 lb. except for Philadelphia & Reading box car, in which the weight used was 24,000 lb. (assumed as two-thirds total weight of car body and underframe). The lading $B$ for 60,000 lb. capacity cars was assumed as 66,000 lb., for 80,000 lb. capacity cars was assumed as 88,000 lb. and for 100,000 lb. capacity cars was assumed as 110,000 lb. The oscillation $C$ was taken at 20 per cent. of the total sum of lading and weight of car body carried.

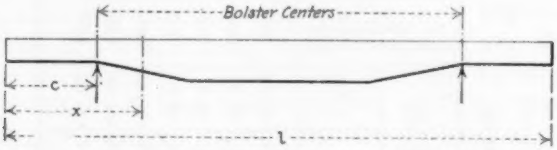



Fig. 29 Diagram for Determining Bending Moments

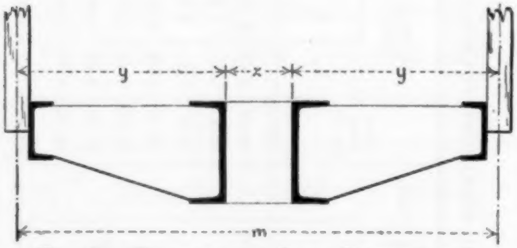



Fig. 30 Diagram for Load Distribution

43 The calculations for center sill load for box cars with wood side frames were made as follows: load per square inch of floor space

$$w = \frac{A + B + C}{lm}$$

Total uniform load on center sills

$$W = wl\,(x + y)$$

where

$l$ = length over centers of end posts, in inches
$m$ = distance over centers of side posts, in inches
$x$ = spacing of center sills
$y$ = distance from back of center sill to center of side post, in inches (see Figs. 29 and 30)

44 The calculations for center sill load for box cars with steel side frames were made as follows: load per square inch of floor space

$$w_1 = \frac{\frac{A}{3}+B+C}{lm}$$

Total uniform load on center sills

$$W = w_1 l\,(x+y)$$

Bending moment of center sills

$$M = \frac{W}{2}\left(\frac{l}{4} - c\right)$$

where

$M$ = maximum bending moment at center of car

$c$ = distance from center line of bolster to center of end post, in inches

$\frac{M}{S_c}$ = compression per square inch

$\frac{M}{S_t}$ = tension per square inch

Bending moment at any point $x$

$$M_x = \frac{W}{2}(x-c) - \frac{Px^2}{2}$$

where

$x$ = distance in inches from center of end post to any section between bolster and center of car

TABLE 15 DATA COMPILED ON AREAS AND STRESSES IN CENTER SILLS

| Area of Center Sills | At Center | | | At Bolster | | |
|---|---|---|---|---|---|---|
| | Min. | Max. | Aver. | Min. | Max. | Aver. |
| 36-ft. cars, wood frame | 26.84 | 45.01 | 33.57 | 23.90 | 36.18 | 26.64 |
| 36-ft. cars, steel frame | 18.73 | 26.05 | 22.37 | 19.80 | 26.05 | 22.90 |
| 40-ft. cars, wood frame | 32.09 | 33.125 | 32.60 | 19.08 | 27.30 | 23.19 |
| 40-ft. cars, steel frame | 26.89 | 35.91 | 30.52 | 22.34 | 30.68 | 25.35 |
| **Maximum Combined Stress** | **At Center** | | | **At Bolster** | | |
| | Min. | Max. | Aver. | Min. | Max. | Aver. |
| 36-ft. cars, wood frame | 12787 | 23519 | 18321 | 10220 | 21124 | 15809 |
| 36-ft. cars, steel frame | 11331 | 20714 | 15380 | 18297 | 22091 | 19671 |
| 40-ft. cars, wood frame | 13772 | 23583 | 18677 | 29975 | 35620 | 32798 |
| 40-ft. cars, steel frame | 16909 | 19831 | 18391 | 8715 | 17713 | 13095 |

$P$ = load per lineal inch of sill

In calculating the Pennsylvania Railroad box car, it was assumed that the cross bearers transmitted to the center sills a certain proportion of weight of the car body, thus producing an additional bending moment in the center sills.

45 From Tables 11 to 14 are obtained the data presented in Table 15. For the 36-ft. cars with wood body framing, it would appear that the averages shown for the area and stress both at the bolster and the center of the car approach figures which are safe for general practice. For the 36-ft. cars with steel body framing, the average stress per square inch behind bolster is too high for safe practice and should be reduced by increasing the area. The low combined stress at center of car is due to the center line of coupler being located about 5 inches below neutral axis, which reduces the compression due to

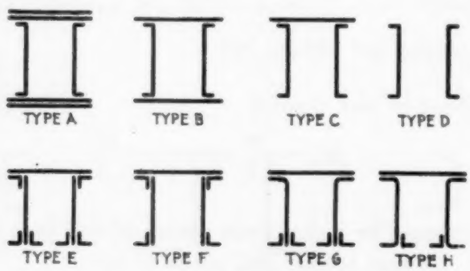



Fig. 31 Typical Sections of Center Sills showing Lack of Standardization

direct end shock. At the center of car the stress due to lading alone is also too high for safe operation. For the 40-ft. cars with wood body framing, the average stress per square inch is entirely too high and should be reduced by increasing the area of the section and cutting down the eccentric load due to end shock; this also applies to the section at the center of the car. In the 40 ft. cars with steel body framing, the maximum combined stress at the center of the car is a trifle high.

46 In all of these cases it is well to note that the relation of the center line of the coupler to the neutral axis has an important bearing on the strength of the sills. At the center of a car with fish-belly type sills the center line of the coupler is usually above the neutral axis, which adds to the total combined compression; whereas, with steel sills, the coupler center line being below the neutral axis, will counteract somewhat the compression due to impact blow. This

leads me to believe that nothing is gained by making the depth of fish-belly sills at the center any greater than is required to take care of the vertical loads. It will also appear that a fish-belly type sill is necessary for all types of cars mentioned of 80,000 and 100,000 lb. capacity, except 36 ft. steel side frame cars, which will render good service when using a center sill construction of plates and channels.

47 *Table* 10. This table shows the general practice in designing body bolsters, crossbearers, side sills, and draft sills. For body

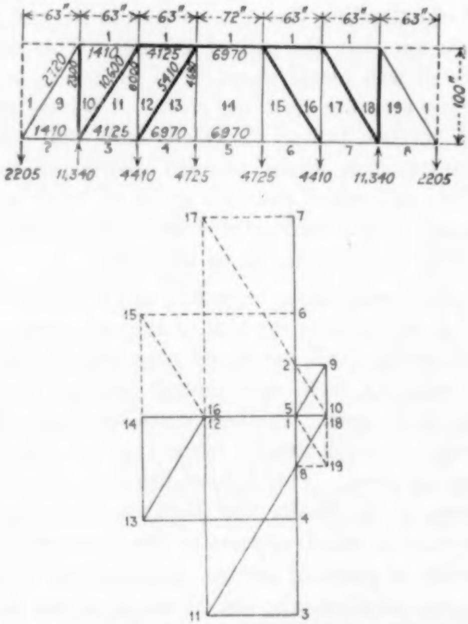



**Fig. 32 Graphical Method of determining Stresses in the Members of Steel Side Frames**

bolsters, it is general practice to use two pressed steel channels spaced at various distances apart with cover plates top and bottom. The best designs have a spacing 6 in. to 8 in. between webs. For crossbearers, a variety of designs exist, the later types consisting of heavy I-beams passing directly through the center sill web plate and extending the width of the car between side sills. This plan is also followed out on the intermediate floor beams and makes a neat and substantial arrangement. The general practice is to use a pressed steel channel with top and bottom cover plate.

48 For draft sills, the practice is, where structural steel channels are used for center sills, to extend the same channel to the end sill to serve as draft sills. It has been customary when fish-belly type center sills are used, to provide a pressed steel Z-shape draft sill and splice to center sill web plate projecting through the bolster. There is a tendency on the part of designers today to do away with the splice by extending the web plates of center sills and providing outside angles to form the draft sill, using a continuous cover plate. Where pressed steel center sills and cover plates are used, the practice has been followed of extending this construction to the end sill, extending the cover plate too near the end of the sill. This construction requires the use of web plates about 5/16 in. thick, so as to provide sufficient bearing area for draft lug rivets. I believe when draft sills have sufficient net area behind the bolster stop, that considerations of economical construction would warrant dispensing with the splice, as in view of the additional cost of a splice on 1000 cars, the expense of its application is not warranted when considering the number of sill failures likely to occur due to its omission.

49 As to draft gears, data regarding the type used have not been tabulated. It is sufficient to say that of the cars enumerated, six were equipped with spring draft gears and nine with friction gears. The impact blow resulting from cars coming together is practically absorbed by the draft gear. Friction gears are more efficient in this respect, absorbing a much greater percentage of total energy as compared with spring gears. This subject has been thoroughly discussed by the members of the Master Car Builders' Association some years ago, a full account of which appears in their proceedings. Owing to the large variety of gears in service, necessitating numerous designs of draft lugs, key attachments, etc., it would appear that, if a standard arrangement of draft gear and all appliances connected therewith were adopted by all railroads, it would result in great economy of maintenance. This should include striking plates and carry irons, which on a large number of cars are not of sufficient strength due to arrangement of end sills.

50 The stresses in steel body framing can be obtained by graphical as well as analytical methods. Fig. 32 indicates the manner in which the stress diagram for vertical loads is obtained, a uniform load of 70 lb. to the lineal inch having been considered, corresponding to a uniform load of approximately 30,000 lb. for a 36-ft. car. The light lines indicate tension and the heavy lines, compression. The side

sill and top plate at the door opening are subjected to the same stress, but it is the usual practice to provide more area in the side sill to take care of unsymmetrical loading. The posts and braces must also resist crossbending stresses due to pressure of lading. The bending moment due to this cause is

$$M = \frac{H^2 \tan^2 \theta D W}{18 \times 1728}$$

in which

$H$ = height of load in inches
$D$ = one-half distance between posts in inches
$W$ = weight of lading in pounds per cubic foot
$\theta$ = ½ (90 deg.—angle of repose)

51 By examining the tables shown giving the comparative data of various items which enter into the construction of a box car, it is surprising that more has not been done in the way of standard construction. The large expenses which railroads are now compelled to face due to repairs of freight cars could be partially reduced if standard designs were in use throughout the country. Repairs would be facilitated due to the use of standard materials throughout for various types of cars, fewer cars would be held up at car repair shops awaiting material from foreign roads, and interchange of cars would not be a hardship to any railroad, as all cars would be of equal strength. Also drawing room expenses would be reduced both for the railroad and car builder, and repair parts could be produced by cheaper methods than as followed out at present, due to elimination of a variety of designs and shapes, principally castings and pressed steel parts.

No. 1405 *b*

# STEEL UPPER FRAME BOX CARS

By R. W. Burnett,[1] Montreal, Canada

Non-Member

While this paper is confined solely to superstructure details of the steel frame box car, it is intended to apply in general to steel frame practice as developed in connection therewith. While the early development of steel upper framing is passed over rather briefly, many of the important considerations that have influenced its adoption are discussed in detail, particularly as viewed by the Canadian Pacific Railway. The information and data presented are based on the writer's experience in this railway system in connection with the design, construction and maintenance of 30,000 cars of this type, which represent an investment of $30,000,000.

2 Credit is due to Mr. C. A. Seley, mechanical engineer of the Rock Island Lines, for designing the first outside sheathed steel superstructure box cars that were constructed in large numbers. The introduction of steel into the superstructure of the box car, and the development of the outside sheathed steel superstructure in particular, were discussed so thoroughly by Mr. Seley in his comprehensive paper [2] before the Franklin Institute in January 1910, that I have thought it unnecessary to go over this same ground, but will review only briefly the development of the box car from the all-wood car through the intermediate stages of steel underframe cars.

3 The original wooden car, with the single spring draft rigging having the check castings bolted to the sills, gave little if any more trouble than modern equipment, due principally to the shorter trains, lesser density of traffic and to the use of link and pin couplers which compelled gentler handling of trains than is prevalent today. The steel underframe car was built mainly to secure a stronger center construction for the attachment of draft rigging and to get away

[1]General Master Car Builder, Canadian Pacific Railway Company.

[2]Vol. 169, No. 4, April, 1910, p. 278.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

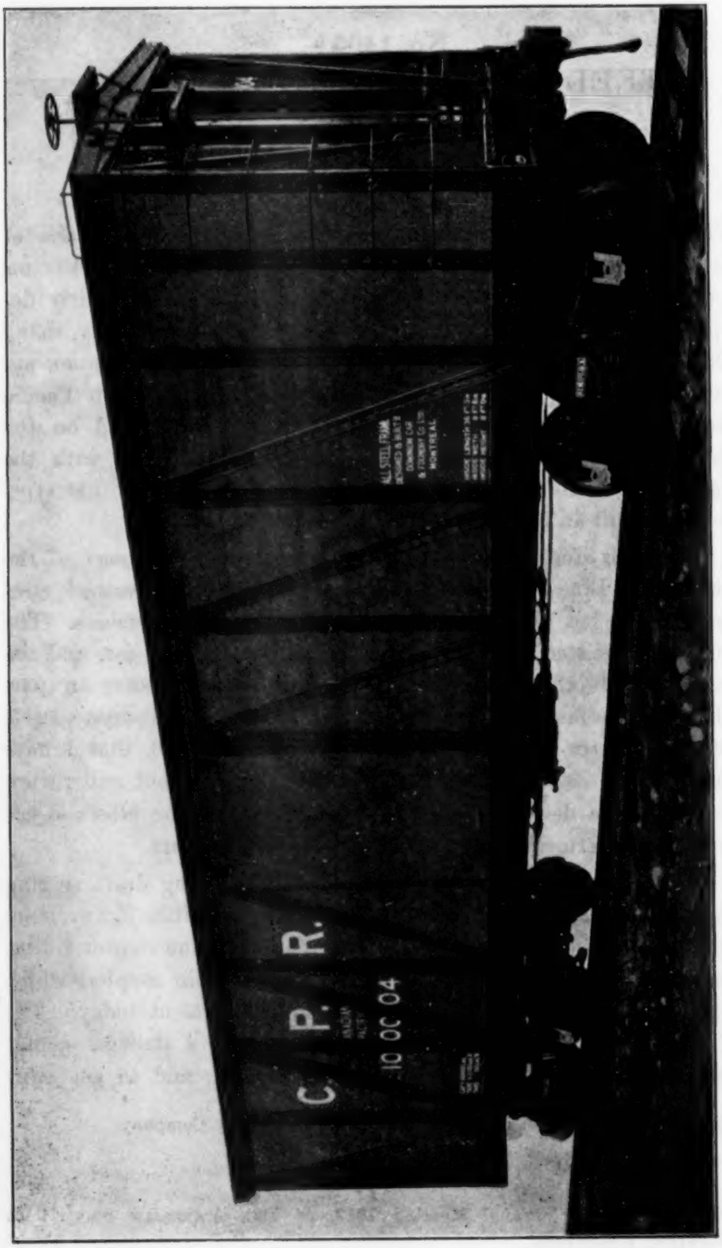



Fig. 1 First Inside Sheathed Steel Frame Car Built by the Canadian Pacific Railway

from the trouble caused by wooden sills breaking and splitting, broken draft bolts, etc.

4 While having many advantages over the old wooden car, the steel underframe car developed some troubles peculiar to itself, the most important being due to the fact that the body being carried on a rigid frame and not held together by the strains resulting from its weight, as in the old trussed cars, has a tendency to develop slack in the superstructure. This in turn affects the roof and sheathing. One principal trouble with outside sheathed cars is that, after they have been in service a comparatively short time, the sheathing frequently loosens at the end sill and at the side sills near the bolsters with resultant leakage of grain.

5 There were some steel frame box cars built previous to 1909, but the writer has been able to secure data only on the outside sheathed types. Of these 2700 were in service on the Norfolk & Western, of which the first 100 were built in 1902; the owners advise they were satisfactory and the same type has been purchased on subsequent orders. The Rock Island and Frisco lines had in service at that date approximately 5000 cars similar to the Norfolk & Western, and these also appear to have given satisfaction as the owners have re-ordered the same type several times. All of these, however, were outside sheathed and as regards leakage at the sills, had comparatively little advantage over the wooden cars. Recently both of these lines have purchased some inside sheathed cars. The Frisco car of this type is fully described in The Railway Age Gazette, October 3, 1913.

6 In 1908 the Canadian Pacific Railway designed the steel frame inside sheathed box car as shown in Fig. 1. This car avoided the disadvantage of the outside sheathed car which had not been accomplished by the steel frame cars constructed up to that time, and at once obtained a further reduction in weight and provided for cheapness of maintenance by the use of steel superstructure, without the additional lumber required by the outside sheathed car. With practically no preliminary experimenting 500 of these cars were built. and since then over 30,000 have been built similar to the first cars, with the exception of several refinements of details, such as corner and door posts, end doors and side plates, and joining of flooring and lining. These changes have not affected the general design of the car, but are improvements that have been introduced from time to time to reduce weight and simplify the construction. The latest type of car is shown in Figs. 3 to 5.


FIG. 2 TYPE OF INSIDE SHEATHED STEEL FRAME CAR USED BY THE ROCK ISLAND SYSTEM (40 FT., 40 TON)

7 The steel frame outside sheathed car has several advantages over the types previously used, notably in that the tare ton weight is low in proportion to the capacity. There is such a variation in the figures used for the cost of hauling per ton-mile, that no attempt is made to say what the saving would amount to, but certainly the advantage of having a car equal, if not superior to other cars in all respects, weighing from 1000 to 5000 lb. less, will appeal to all traffic and operating men. Not only is there that much less dead weight to haul when the car is empty or partly loaded but additional lading can frequently be carried. The actual limit on the paying load that can be carried in a properly designed car is the total weight on the axles. Thus, a car having 5 in. by 9 in. axles with such a tare weight that, when deducted from the capacity of the axles, it allows the car to be safely loaded to 88,000 lb., could, if dead weight be reduced by 3000 lb., safely carry a paying load of 91,000 lb. and retain the same strength. Thus the actual capacity of the car is increased almost 4 per cent with a better ratio of paying to dead load.

8 With the wooden superstructure, it had been thought necessary to assist the superstructure by heavy roof construction, some going so far as to use different methods of diagonal bracing, but with the steel car it has been found that there is no appreciable local movement of the framing in the heaviest service which makes a simple proposition of the roof as it has only to take care of itself. This presents a simpler problem to roof designers, making it possible to design a roof much lighter, without necessity for use of purlins or ridge poles to strengthen the car. It is obvious that unnecessary weight in the roof raises the center of gravity, and increases the tare weight and cost and has other disadvantages.

9 In explanation of the local movement of this style of framing, it is well to mention tests we have made in jacking up this car, which demonstrated that the car would take a gentle twist from end to end, allowing the bolsters to be slightly out of the same plane horizontally. This twisting was accomplished without any perceptible local distortion of the sides or ends. The capacity for twisting is a condition to be desired as it allows a car to adjust itself to uneven track conditions.

10 In addition to being $5\frac{1}{2}$ in. narrower than the outside of the sheathing of a wooden car, the superstructure of the Canadian Pacific Railway car is protected by the framing, so that a side swipe that would do serious damage to an outside sheathed car frequently does not touch the lining and is resisted by the framing without damage

FIG. 3 STEEL FRAME OF CANADIAN PACIFIC RAILWAY BOX CAR READY FOR LINING

to the posts or braces. Frequently it is found that a side swipe that would almost demolish the sides of a wooden car only bends the steel framing, and in making repairs, the lining is merely removed, posts and braces straightened and the original lining replaced, the whole cost being the comparatively small labor charge. Jacking frames are being installed at all of our principal repair points for all classes of steel cars, and while not original with the Canadian Pacific Railway, have been amplified better to take care of steel frame box cars. With these frames, many jobs that would require the car to be cut apart, taking several days, can be done in a few hours without cutting rivets. With modern steel frame cars, these jacking frames are as much a necessity as the blacksmith shop or any other part of the shop.


FIG. 4 END VIEW OF CANADIAN PACIFIC INSIDE SHEATHED CAR SHOWING GRAIN-TIGHT END DOORS

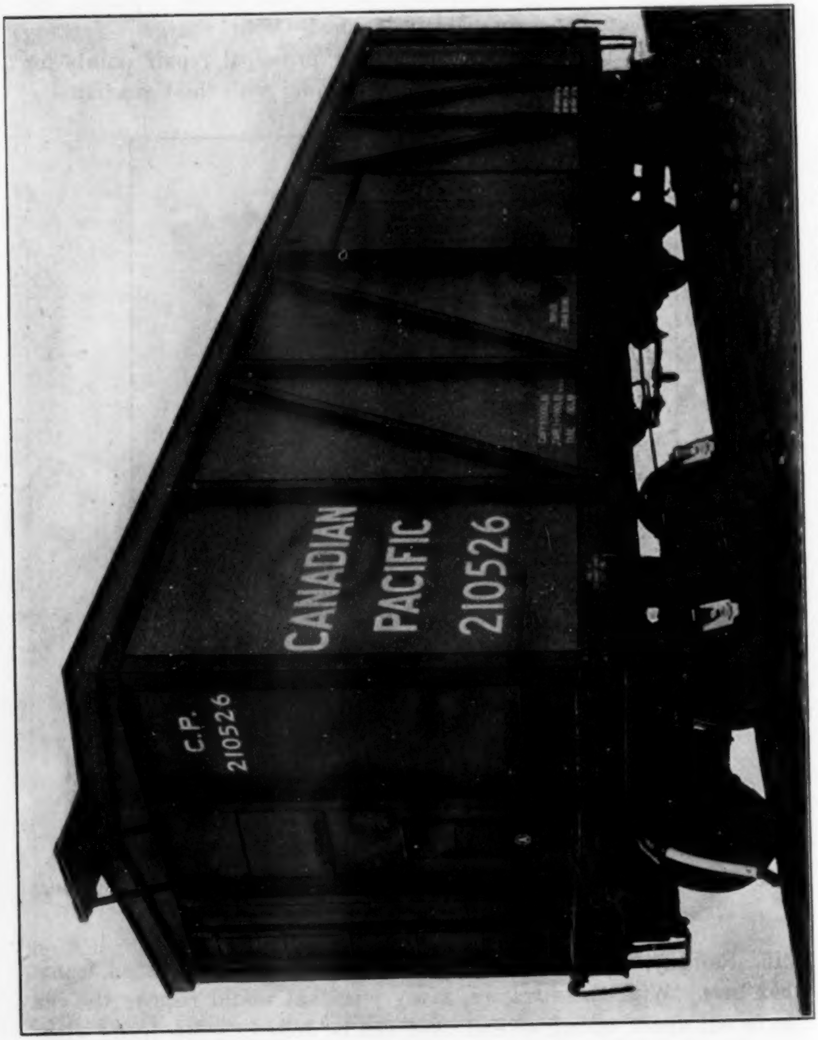



Fig. 5 View of One of the Latest Inside Sheathed Cars Built by the Canadian Pacific Railway (36 Ft., 40 Ton)

11 With the outside sheathed car, it is difficult to clean a car properly when it is unloaded, on account of grain lodging between the framework and also on account of the opening where the posts and braces meet at the bottom becoming obstructed, resulting in grain being retained between the sheathing and lining with consequent complaints from shippers. All of this is overcome by the clean join-

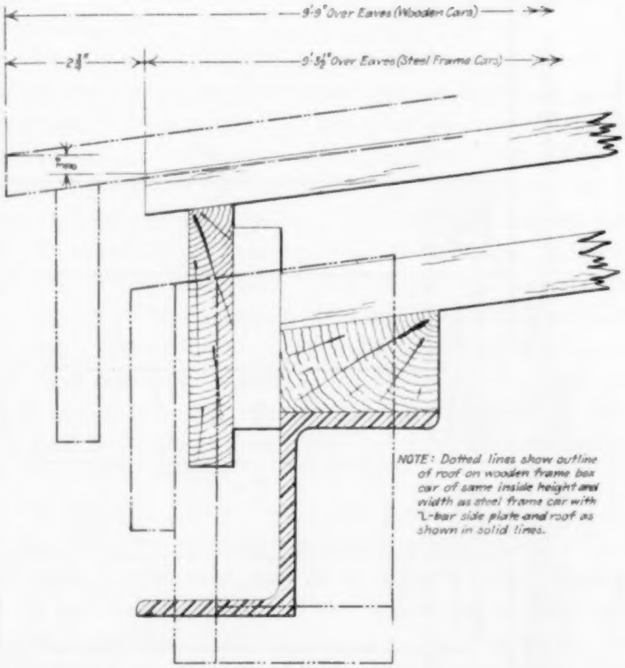



Fig. 6 Detail of Roof Clearances on Steel Frame and Wooden Box Cars

ing of the lining and the floor in the steel frame cars, and it is believed a change of this kind would have come years sooner if designers had kept in close touch with service conditions. One advantage of the steel frame car is that outside of possible repairs due to wreck damage and to wear and tear of couplers, wheels, brake shoes and journal bearings, the car does not deteriorate more rapidly in service than when stored.

12 The grading of lumber for use in these cars is an item that has received much consideration. Yellow pine or fir has so far been

REPORT OF MOISTURE IN LUMBER FOR LINING BOX CARS

Built by.......................................................At.......................................................................

C. P. R. Equip. Order No......................For.................................................Number of Cars......................

Report No.....................

Moisture determination strictly in accordance with C. P. R. Spec. No. 243 C.

Samples obtained every other day during construction, with a minimum of one sample for each one hundred cars built. Result of each test must be promptly recorded on this sheet and sent to R. W. Burnett, Genl. M. C. B., Montreal

| Lab. No. | Date of Sample | Cars Completed | Car Nos. Represented | Moisture Per Cent | Average To Date | Orig. Width Inches | Shrinkage Inches | Average To Date Inches | REMARKS |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | |

Signed

Engineer of Tests.

Fig. 7 Form used for Reports to Canadian Pacific Railway on Moisture Determinations on Lumber

the principal lumber used, although we have experimented to some extent with spruce. Spruce has the advantage of being lighter, but it seems to be more difficult to dry it sufficiently for this purpose. Great pains have been taken to avoid knots that are too large or numerous and while it is generally desirable to have lumber as free from knots as possible, I have never in the inspection of many hundreds of cars, seen where a knot had fallen out. It is, however, desirable to have lumber as free from sap and shakes as possible and thoroughly dry.

13 When the first of these cars was built outside of the Canadian Pacific Railway shops we had considerable difficulty in getting the lumber properly dried due to lack both of experience and facilities on the part of the car companies. We have about 3000 cars on which the lumber has shrunk, giving them a bad appearance, but this result was expected, as when the cars were built the lumber was quite green. The sheathing on these cars could be tightened for less than $4 per car, but very few have been tightened owing to receipt of practically no reports of loss or damage to lading due to the shrinkage; also as they do not frequently reach our main repair tracks, being shopped only for such repairs as wheels, or wreck damage, we have not considered it advisable to shop the cars for a defect which is almost entirely a matter of appearance. The lining shrinks as much in two months of summer weather, as it ever will.

14 The lining should not be matched before drying, as it warps and curls, rendering it difficult to make a tight joint. The rough size of lumber should be at least $\frac{1}{4}$ in. greater than finished dimensions. In establishing limits for drying lumber no information or data could be secured whatever, and after experimenting we came to the conclusion that a piece of this lining of full cross-section subjected to a temperature averaging 170 deg. fahr. for 96 hours should not lose more than 6 per cent in weight and that lumber represented by samples losing more than 10 per cent must not be used until further dried.

15 The variable condition of the lumber when taken from the yard makes it necessary to use careful judgment as to the length of time it should be kept in the kiln. At the Angus shops of the Canadian Pacific Railway this responsibility falls on the wood-mill foreman whose constant attention to this feature makes him the best fitted for the purpose. The average moisture loss reported by


FIG. 8 VIEW OF STANDARD CANADIAN PACIFIC STEEL FRAME CAR FITTED WITH CORRUGATED STEEL LINING

the test department for lumber used on cars now building at the Angus shops is 5.25 per cent which shows that we are getting very satisfactory results from the kilns. A number of tests were made last year on lumber taken from the yard. These tests showed a moisture loss of between 25 and 30 per cent which shows the importance of drying lumber properly. The form shown in Fig. 7 is used for reporting results of tests both at Angus and outside shops.

16 Due largely to our insistence, nearly all the car plants in the country are now equipped with dry kilns, and any possible

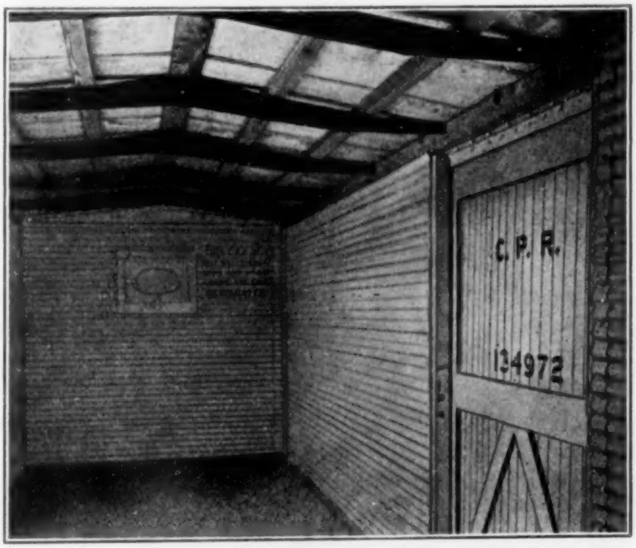



FIG. 9 INTERIOR OF CANADIAN PACIFIC STEEL FRAME CAR WITH CORRUGATED STEEL LINING

additional cost of drying lumber, in excess of what has been considered good practice in the past, would be less than $1 per car. Such drying would make the car side practically the same as one board so that it is absurd that the possible shrinkage of lumber should be considered as any reason for this type of car not being built. It has been claimed that lumber can be so dried that it will swell and bulge, but we have never found this to occur. We have had cases where lumber slightly moist has dried more rapidly on the inside, due to that side not being painted, and made the outside of the boards slightly convex, with tight joints that could be easily mistaken for swelling, whereas the opposite is the case. We have

kept a car with very green lumber in the passenger shop with a high temperature for over a month until the lumber was absolutely bone dry, and then put it outside with doors open through four weeks of constant raining spring weather, with the result that there was no closing of the cracks that could be detected, which further proves that there is nothing to be feared from lumber being too dry.

17 The defects in the sheathing that must be most closely watched are shakes or splits that extend obliquely downward into the

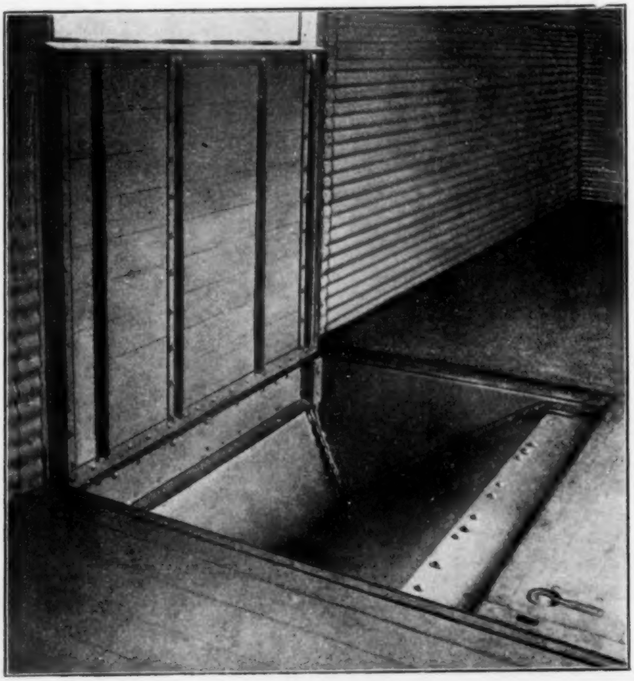


FIG. 10 DETAIL VIEW OF HOPPER OPENING IN FLOOR OF CORRUGATED STEEL LINED CAR

car which must be knifed in with paste before the car is painted. The edges of the lining should be painted, and we have found this can be done more easily and thoroughly by dipping the boards and putting them through between two rubber scrapers which removes the surplus paint leaving the edge thoroughly coated. This gives a thin coat of paint on the inside of the car which is an advantage, causing the lumber to dry more uniformly and diminishing the tendency to warp. Narrow boards have the advantage of having

less tendency to warp, and also if the lumber should not be thoroughly dry, there is less total shrinkage for each board making the space between the edges narrower. The steel work and roofing are painted the same as other cars. This considerable space has been given to the grading and drying, and painting of lumber, as we have found that these factors have required much more attention than everything else combined in connection with the car.

18 The development of the inside sheathed car has been so rapid and the experience with it so uniformly satisfactory, that I feel safe in saying that its introduction in such large numbers on so many roads in so short a time indicates more nearly a tendency toward the adoption of a standard car than has any distinct type of car outside of patented cars for special service. It is certain that there will be no backward movement to a wooden superstructure, and that this car with possible modifications will remain a standard car unless some superior type of car is developed. It may be stated as the writer's opinion that no committee will ever develop a car that will be adopted as standard but that the nearest we will ever get to a standard is what may be developed by one or two persons given a free hand, the merit of which is so pronounced that it forces itself upon the country.

19 With the use of structural steel there is less necessity of carrying special parts in stock on account of repairs being largely a question of labor, and it seems that with this type of car the necessity from a repair standpoint for a standard car is decreasing. This is further borne out by the fact that for the 30,000 cars of this type we have ordered no material for repairs and carry none in stock outside of material common to all cars, except lining; of the lining, our stock amounts to practically nothing. We save out sufficient of the parts from cars destroyed to make up our stock of repair parts, but have found it necessary to use very little of this. There are, of course, many valid reasons why cars should be made to standard inside dimensions and outside clearances.

20 To look at the matter in another way, the wheels, axles, journal bearings, journal boxes, couplers, brakes, safety appliances, etc., which constitute the removable and perishable parts, are all standard and when it is remembered that nearly all of the remaining parts of the cars are standard rolled shapes which are easily obtained either from the mill or from stock in all principal cities, it is apparent that we now have, in effect, a standard car, or at least a car of standard parts. A car of different dimensions would not increase

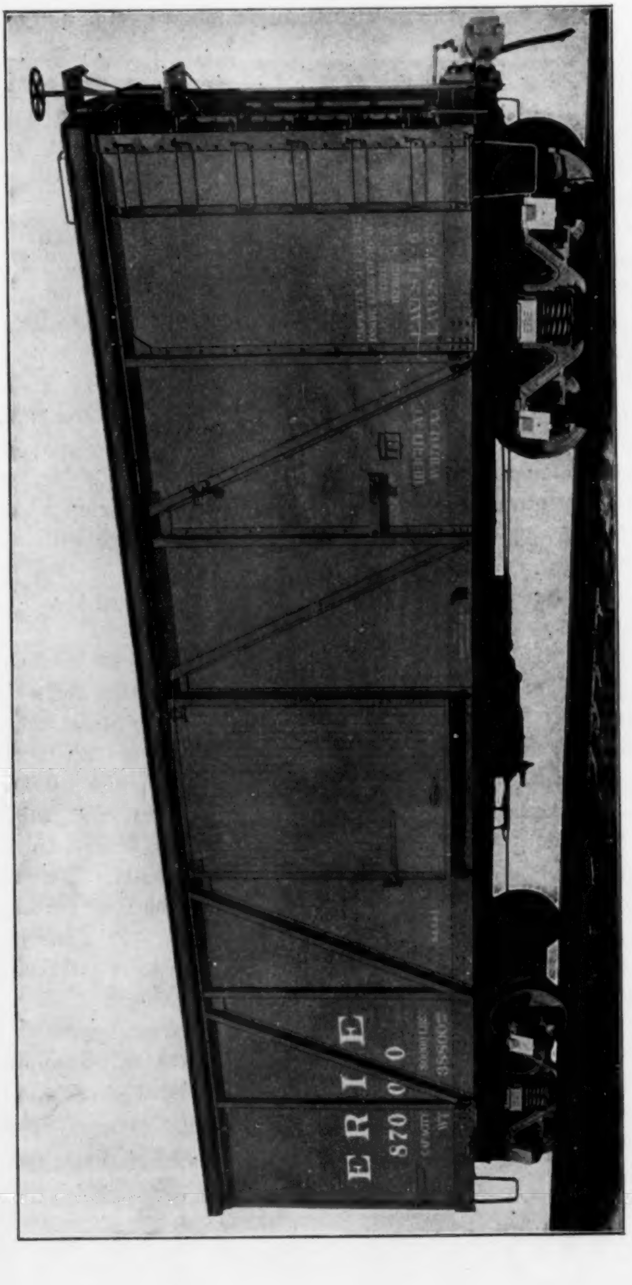



FIG. 11 VIEW OF STEEL FRAME INSIDE SHEATHED BOX CAR IN USE ON THE ERIE RAILROAD (36 FT., 40 TON)

the cost of maintenance as long as standard shapes are used; nor would it if every lot of cars is designed differently, as long as proper strength is maintained, and any change in design would usually be to increase the strength. In other words, to keep a car as close as possible to standard and reduce cost of maintenance, rolled shapes should be used in preference to pressed shapes where possible.

21 It is my belief that the people who are urging the adoption of a standard car for maintenance reasons have in mind the remaining wooden cars for the maintenance of which large quantities of timbers and castings have to be kept in stock. It is of vital importance that the parts be standardized if that style of construction were to be continued. It should not be overlooked that in a car constructed with rolled shapes, these parts seldom need renewal even when the car is wrecked, as they can easily be straightened or formed to the original shape at any car repair point, while wood would have to be replaced and pressed shapes would call for special dies to reform them. With a wooden box car the amount of material necessary to carry in stock and use for repairs increases rapidly with the age of the car. With a steel frame box car the amount of material necessary to carry in stock and use for repairs outside of parts common to all cars does not increase with the life of the car.

22 The wind resistance on the steel frame box cars with inside sheathing is slightly greater than on a smooth outside sheathed car, but on the other hand, it is less than on any ordinary type of stock car. The effect of wind resistance between box and stock cars, has never been great enough to require any distinction between them as to the number of cars that could be hauled in a train of either and is really a refinement that not even a dynamometer car can detect. A small change in the angle or velocity of the wind, or difference of the number of wheels running to one flange, or trucks somewhat out of square, affects the haulage of the train too much to enable any satisfactory figure for the difference in the wind resistance of the various types of car to be obtained. There is a certain stretch of track on the western plains of about 40 miles without a curve and practically level where high winds are frequent, on which the haulage capacity of locomotives is dependent principally upon the wind, and yet even there it was found practically impossible to distinguish between the wind resistance of stock and box cars. From this, it is evident that the wind resistance of steel frame inside sheathed box cars as compared with outside sheathed cars may properly be ignored.

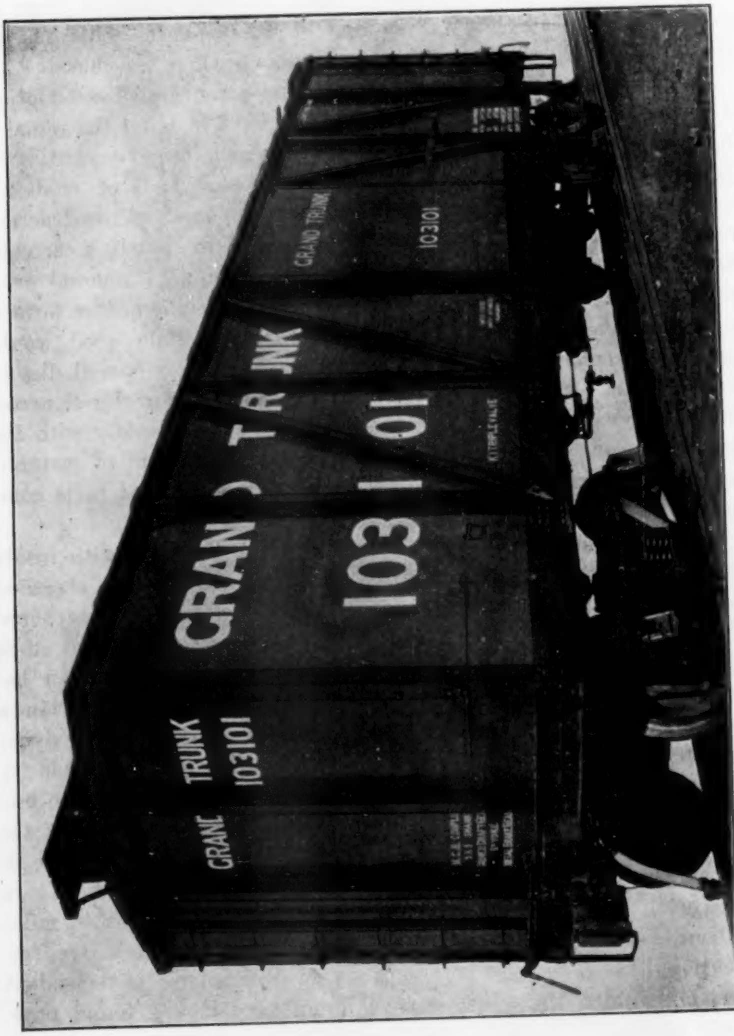



FIG. 12 STANDARD STEEL FRAME BOX CAR WITH METAL ROOF IN USE BY THE GRAND TRUNK SYSTEM (36 FT., 40 TON)

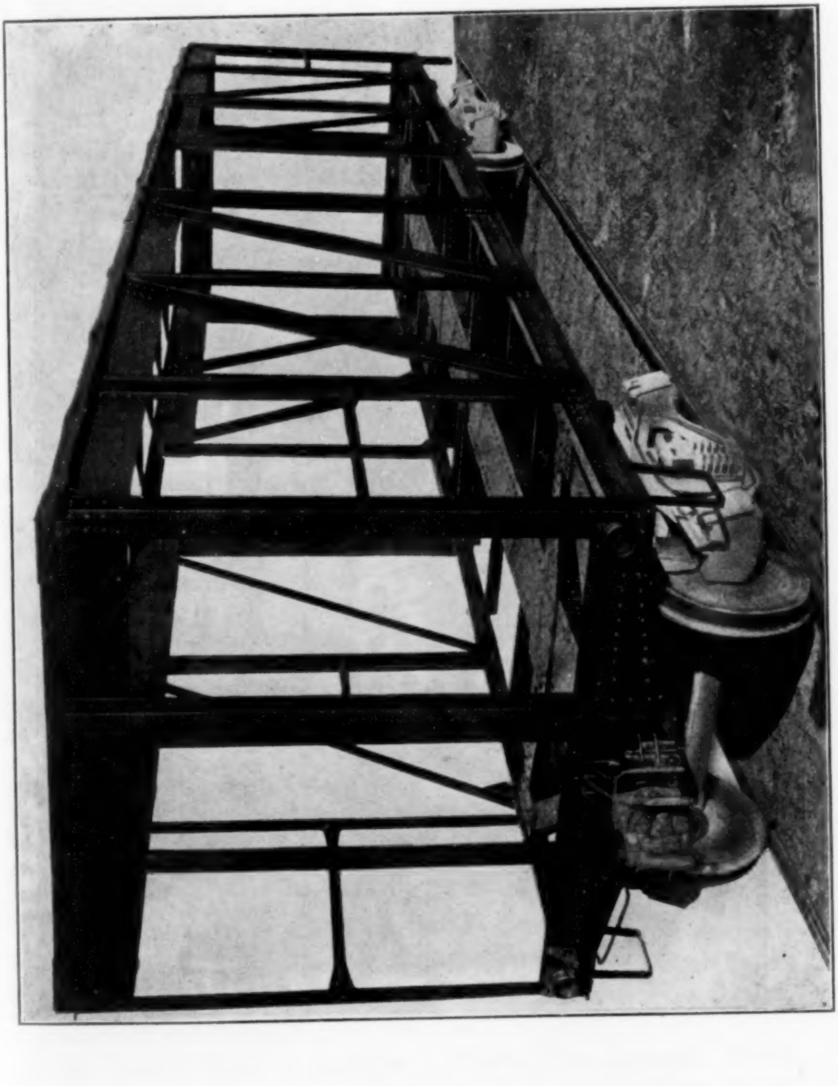


FIG. 13 VIEW OF STEEL FRAME OF THE ROCK ISLAND CAR (FIG. 2) READY FOR SHEATHING


Fig. 14 Inside Sheathed Steel Frame Car used by the Intercolonial Railway (36 Ft., 30 Ton)

23 In the summer of 1911, we lined one of these steel frame cars with corrugated steel and found it to be as simple a matter as lining with wood, Figs. 8 and 9. We lapped and rivetted the sheets, which were No. 13 gage, between the door and end, and had the corrugations on the side and end coincide, pressing into special corrugated angles in the corners to break the joints. At the floor, we straightened out about 4 in. of the corrugation and formed of it an angle that rests on the side sill, and on this the ends of the floor boards were superimposed, easily making as tight a joint as I have ever seen on any car. After 18 months of general service this car was brought in and on examination, found to be in as good shape as when constructed. It was interesting to note that, when inspected, the paint sealing the joints, where the side sheets lapped, was in no place seal broken, indicating that there is no material weaving or deflection of the sides. The paint was in perfect condition, there still being some gloss, indicating that in the use of steel there is no disadvantage as far as the painting is concerned. Different methods of lining with steel could be followed, and I am convinced that if experience proves that there is no damage to be feared from heat, cold or sweating, that steel lining will be largely used. But, I am also convinced that the use of steel lining with any insulation will never be extensively used as it adds to the cost and weight without affording any protection to the lading, which is not secured by the wood lining. An advantage of this construction is shown in Fig. 10, in the application of hoppers under the door openings, which were made without alterations to sills or cross bearers.

24 As regards the end of the car, Fig. 4, we use two 4-in. Z-bar end posts of 8.2 lb. per ft., with 1¾-in. lining which gives good service, but we intend to use on future cars two 5-in. end posts of 11.6 lb. per ft. with 2⅜-in. lining for a height of 4 ft. and 1¾-in. lining above that height. This, we feel, will protect any lading that needs protection. If a car gets such rough handling that wheels or rails, or similar lading would break through, it is better to have the boards broken than to distort the posts, as the lining can be replaced at any repair track with a minimum expense, while distorted posts would require sending the car to a steel car repair point. The single thickness end lining makes convenient the application of single thickness, grain-tight end doors.

25 Out of 30,000 of these cars, 29 have been destroyed. Based on the length of time in service, this would average a loss of approximately one car per 1000 per year. Of the cars destroyed, 15

were burned, 14 were destroyed in wreck, 10 cars being destroyed on foreign lines. As the loss of cars by fire is in no way affected by the details of construction, I will eliminate them from the calculations. This then, based on the length of time in service would give about one-half car per 1000 per year destroyed in wreck. As there is no appreciable deterioration of these cars in service, it is safe to assume that in the same service substantially the same rate of loss would continue, while with wooden cars the rate of loss would increase each year.

26 A conservative estimate shows that there are today approximately 65,000 steel superstructure cars, including outside sheathed, in service. Of this number 30,000 are Canadian Pacific Railway and nearly all the remainder belong to the roads whose names appear on the cars illustrated herewith. These I am able to show through the courtesy of the officers in charge of the mechanical departments of the several lines, whom I wish to thank for their willing assistance.

No. 1405 c

# DISCUSSION ON STEEL FRAME BOX CARS

H. H. Vaughan thought that Mr. Burnett's position was sound, that the structural steel car was to all intents and purposes a standard, as far as any car could be standard. He believed that one standard type or design of car would never be adopted. There were sure to be improvements and alterations that the different roads might think desirable; if standard material, easily obtainable, were used, and if certain parts, mentioned by both Mr. Rink and Mr. Burnett, were kept standard, that would be as close to the standard car as would ever be approached.

The draft castings, arch bars, bolsters and some of the other parts should be standardized to a greater extent than at present. It was absurd that the slight variations in these parts necessitated their being obtained from the car owners when repairs were made on foreign lines.

Between the bolsters and the sills, the cars were practically unsupported except for side bracing. Mr. Vaughan stated he would not advocate this construction if the Canadian Pacific cars were running in heavy coal trains or under such conditions as the Norfolk & Western have in their coal service. In designing a car it was necessary to figure on the service the car was generally going to run in, not the service it might run in. It was felt that 60 per cent to 75 per cent of the service to which the box cars were put, both in Canada and the United States, was service in which the old type of underframe would stand up satisfactorily. These figures were justified by the result obtained with this type of car, in a period of five years, as given in Mr. Burnett's figures.

As far as was known, not a single car had shown vertical weakness in the center sills. The omission of that cover plate had introduced a certain amount of longitudinal weakness through the center, as well as buckling sidewise, but in no case buckling vertically. It was thought that the floor would be sufficiently stiff to prevent any lateral buckling of the center sill, and some floors were so loose that they could not have acted that way if the car had been permitted to buckle.

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

The center sills and the side sills had ample strength to hold up the corners of the cars under general conditions, and the saving of 500 lb. of weight there, 500 lb. in the cover plate, and a few hundred pounds here and there, was what had made so light a car. The company had a car weighing 36,500 lb., carrying 40 tons, which could be loaded to 93,000 lb., permissible loading.

Mr. Vaughan called attention to the advantage of reduced weight on net earnings, and not on the cost per ton-mile; the company's figures for 1913 showed that 22.34 tons per loaded car-mile were carried, and that the percentage of light car mileage was 28.5 per cent of the loaded car mileage, which gave an average load of 16.8 tons per car-mile total. The weight of the average car unloaded was 18 tons, so that the average weight of car loaded was 34.8 tons. That was not a bad figure for a large number of roads in this country handling general traffic. The Canadian Pacific handled a large amount of grain, and while it did not have much coal, the average load was fairly good.

Assuming this average car weighed 1 ton more, to begin with, without any question as to whether the train rating would always be changed or not, the weight of the car loaded would go up to 35.8 instead of 34.8; in other words there would be an increase of the ton mileage of 2.85 per cent; or if operating on a ratio of 70 per cent, under this changed condition of weight, the operating ratio would be 72 per cent instead of 70 per cent. The net earnings would drop from 30 per cent to 28 per cent, which was a difference of 6 per cent in net earnings.

The question of weight was an important one in car design. A lighter car might cost \$5 or \$10 more a year to keep up than a heavier one, but it would save two or three times that in the weight to be hauled uselessly about.

Mr. Vaughan agreed with Mr. Rink in the question of sheathing. The Canadian Pacific owned a number of cars which had shrunk considerably, but there had been few cases of damage cases due to them. It was not a difficult matter to tighten them, as Mr. Burnett said, and the only reason why this had not been done was because the company had not received a sufficient number of complaints to justify taking the cars out of service and doing the work. The vertical sheathing would be, possibly, a preferable arrangement if it could be accompanied with an economical and convenient design of side framing. But this would be difficult to accomplish for the truss

form of side framing naturally lent itself to horizontal sheathing. In vertical sheathing it would have been necessary to introduce horizontal members to take care of the fastenings, as in the Pennsylvania Railroad cars. The distance from the top plate to the sill was too great to permit the side sheathing to be placed vertically without any support.

Mr. Rink referred to end reinforcement repairs. The Canadian Pacific Railway were using very extensively the same style end as was used on the steel frame car for repairs of all wooden box cars: it simply put in two Z-bars and reinforced the ordinary end post, and put the 1¾-in. lining inside.

He thought the roof question was the next to come up for discussion: the three types described were at present the most widely used. The Canadian Pacific had been an advocate of the inside metal roof for a long time, though they had experimented with the all metal roof, and would be interested to know what results were being obtained.

Mr. Vaughan in discussing the history and the development of this type of car, said he felt that he was correct in saying that this type had been originated on the Canadian Pacific Railway when W. E. Fowler was General Master Car Builder. The car was the result of a conversation between Mr. Fowler and Mr. Vaughan relative to policy in turning toward steel underframes.

B. D. Lockwood[1] stated that if Mr. Rink's recommendations advocating standardization of cars were to be followed a committee should be appointed by the Society to design such standards, indicating in their work the minimum of strength and empty weight and the reasons therefor that should be acceptable from a railroad standpoint, so that car builders could govern their action properly in competition. The joint efforts of the railroad committee of the Master Car Builders' Association and the United States Railway Mail Service in connection with the standardization of postal cars resulted in a series of recommendations, and the adoption afterward, of standard specifications, in which the minimum acceptable section modulus was given as the governing factor for all principal members and there was no reason why this could not be done in the case of recommended standards for freight cars.

Mr. Rink had outlined a system of calculations, but whether his deductions were to be considered as generally expressing the ideas of

[1]Ch. Engr., Pressed Steel Car Co., Farmers Bank Bldg., Pittsburgh, Pa.

railroad mechanical engineers throughout the country was a question, and the whole matter should properly be referred to a committee.

H. Wade Hibbard called particular attention to the necessity of designers keeping in close touch with service conditions. A number of years ago he was connected with a large trunk line, where it was the declared policy in the mechanical department that the designers should go frequently into the shops and into the repair yards and into the scrap yards for the purpose of studying how the equipment had failed in service. Later the policy was changed and the men were not expected to carry on that sort of close contact with the service conditions, with the result that locomotives that were found at the main shops to be bad in design for upkeep were re-ordered from the locomotive builders with the same old mistaken design. The policy of one of the large steel car manufacturers of the country was a good one: they had a college educated man go into the shops and learn the business completely; he was afterwards kept traveling among the railroads that purchased steel cars from his company for the sole purpose of learning the good and bad things about these cars in actual service, reporting the result of his efforts to his company.

W. F. Kiesel, Jr. (written). There is one feature in the calculations for end strength to which Mr. Rink has not especially called attention, and that is the ratio of stress to strain under end shock. The important point to consider is that at the bolster. This ratio in the 14 cars enumerated varies between 0.033 and 0.118. The high figures are due to the great distance between the center line of the drawbar and the neutral axis of the sills. The area of the center sills at the bolster varied from 19.8 to 36.18, and the average is 24.75. Without using any additional metal in the center sills the ratio of stress to strain can readily be reduced in many of these cars to the advantage of the strength of the car.

Both Messrs. Burnett and Rink seem to favor the Z-bar posts and braces because they are made of rolled material, and, as stated by them, can be readily obtained. This does not seem to be a good argument, as it is well known that standard sections of rolled material cannot always be obtained on short notice; in fact, within the past year, the steel mills have quite frequently reported that certain angles, I-beams, etc., could not be furnished in less than three or six months.

The four following questions seem important: First, is the box car with steel side frames all that it should be? It has been argued

that shrinkage of lining, driving of nails in lining to secure blocking, breakage of tongues and grooves, etc., will cause leakage, necessitating constant repairs, and that the car with steel sheathing and wood lining is closer to the ideal in box car construction.

Second, in box cars without sheathing, shall all posts and braces be made of rolled steel, or all pressed steel? Advocates of pressed steel assert that pressed posts and braces are lighter per unit of strength, because they can be formed to the required shape; that they can be formed with sufficient surface at the ends for the number of rivets required to develop their full strength, while Z-bars and other rolled forms require gusset plates for this purpose; that they are not likely to be damaged by push-poles, and, if damaged in wrecks, can be readily straightened and restored to approximate shape; that when absolutely necessary to replace them they can readily be obtained from car owner or builder, and that it will not be necessary to wait for any special rolling of material.

Third, should not the posts and braces be considered strictly as beams supported at top and bottom, in combination with straight tension and compression, as members of the side truss? Mr. Rink indicated that flattening of pressed posts and braces, where they connect with side sills, has a weakening effect, which further indicates that he considers them as cantilevers held in vertical position by the side sills and frame braces connected thereto. In wooden cars the posts and braces were strictly beams, and not cantilevers, as they rested on top of the side sills, either directly or on castings with shallow pockets. Side sills of box cars have too little resistance against torsion to hold the posts and braces vertical; they, therefore, must depend on the strength of the side plate and the tying effect of the carline. If, in addition to this, a solidly riveted roof is used, the tops of the posts and braces are securely held in proper alignment and the stability of the side truss is assured.

Fourth, is it not imperative to use diagonal braces in the end framing? No argument need be presented here for this, as Mr. Rink has already furnished sufficient argument, and we know of nothing to show the contrary.

It should be noted that all of the 14 cars enumerated by Mr. Rink have so-called box-girder center sills, and that the majority of them have a minimum section of about 24 sq. in. With this section area a ratio of stress to strain of 0.06 can be obtained, provided proper adjustment is made for relative location of neutral axis of center

sills and center line of draft gear. It would, therefore, seem that the present designs of box cars corroborate the recommendations of the Committee on Car Construction of the Master Car Builders' Association, and that those recommendations are reasonable and conservative. A thorough knowledge of cars by the motive power officials of railroad companies will, we hope, lead them ultimately to endorse the M. C. B. recommendations referred to.

E. G. Chenowith[1] (written). Before a standard for the box car is reached, it appears to me that three very important items entering into the problem must be decided upon in common by all railroads, viz: (*a*) capacity, (*b*) dimensions and (*c*) design. The railroad companies are far from agreeing on either one of these items, as can be seen from Messrs. Rink's and Burnett's papers. One need only stop to consider what an enormous saving there would be in maintenance if there was only one design of car to repair, and what a decrease would result in the enormous quantity of material which is now carried if our stores need carry repairs only, say of a 40-ft., 40-ton steel frame single-sheathed box car. Under this ideal condition our bad-order cars would be on the repair tracks 50 per cent less time.

The design of a car is influenced by many local conditions, and often by a great many local instructions. It is regretted that the merit of a design of a car is too often inversely proportioned to the final weight of the car. I believe that we are now about at the minimum limit relative to weight of box cars, and the tendency is to increase, and not to worry so much about the extra dead weight hauled but to give more consideration to keeping the car in revenue service more days of its life.

Steel underframes are now being applied to thousands of cars in this country and perhaps in as many designs. Whether they should carry a vertical load, or only withstand the buffing, seems to be a prevailing question. Many underframes are being applied to cars retaining the original truss rods, while in many other cases the depth of the steel underframe is made to carry the vertical load, thus eliminating the truss rods, it being claimed that the cost of maintenance of rods will offset the cost of deeper center sills. It must be conceded that in a great majority of cases where any design of underframe is applied, the wooden underframe is not disturbed, so that, therefore,

[1]Mechanical Engineer, Rock Island Lines, Chicago, Ill.

the weight of the car is increased to an amount equal to the added steel underframe.

The main object in view in applying steel underframes is, I think, to obtain a greater resistance capacity for buffing. The car perhaps has with the aid of truss rods properly carried the vertical load, therefore, why not add to the underframe only that which will take care of the buffing strains. To my knowledge, underframes are being applied which consist of 8 in. by 8 in. H-girder beams. These girder beams set up between the wooden center sills, supporting them by means of angle brackets riveted to the flanges. To these girders the ½-in. draft plates are riveted extending from end sill to bolster. As this design was not intended to carry a vertical load, the bolster and cross ties were not connected to the girder in a manner to carry the load imposed on the side sills. Another cheap design of an underframe of this class lies in the use of two 9-in. channels with cover plate; this design sets below the wood sills, however, and therefore requires a bolster and cross tie in order properly to apply truss rods.

In the design of steel underframe consisting of two center members, it seems to be the general tendency to apply top cover plates instead of relying on the diagonal bracing to side sill to keep the center sill from distorting in very severe service.

Referring to Table 2, Mr. Rink states that it is the whim of the designer that occasions so many different lengths of box cars. In my experience in designing equipment for several different railway companies, the consideration of a proper length car was held as a serious question and good reasons were developed in the freight traffic department before the designer received instructions as to the principal dimensions around which to design a car. The popular dimension, it seems to me, for inside length of box car is 40 ft. I know of a railroad operating over 12,000 cars of this length and they seem to meet the requirements very well.

The standardization of design of box cars will have a tendency to decrease the great variation of weights. I saw two designs of cars built within a year of each other by two railway companies, one a 60,000 lb. capacity car having a light weight of 48,000 lb., while the other, an 80,000 lb. capacity car, had a light weight of 36,000 lb., and moreover, both railway companies were perfectly satisfied with their designs.

In the design of steel superstructure cars, I am of the opinion that the side sills should be only of a proper section to complete the

trussed panels of superstructure, and should not be designed as a load carrying member. This will allow the superstructure more flexibility to adjust itself to irregularities of track, and eliminate the tendency to derail. A car held rigid so that the planes of the sides are always parallel will not properly take a curve.

The trouble experienced from water following beading or grooves in horizontal sheathing and then passing into the car, could, I think, readily be overcome by bevelling off the top corner of each board outside of the tongue, which would eliminate the gutter effect at points of matching where the sheathing is not entirely tight. Where single sheathing is used the $1\frac{1}{2}$ in. thickness seems to meet the requirements, and it should be tongued and grooved instead of shiplapped. Some designers are using $1\frac{3}{4}$ in. or even thicker for end sheathing, but the $1\frac{1}{2}$ in. thickness on both sides and ends with extra reinforcement on ends furnished by proper design and location of end posts seems preferable.

The proper method of designing the end of a box car or reinforcing it for shifting loads is a question all designers have given extra time and thought. The end posts of a car are of no consequence except in so far as the end plates, of whatever construction, can properly support the upper end of the posts. This evidently has been overlooked in some cases, as cars will be found having heavy steel end posts attached at top to a light wooden end plate. The end of the car should be so constructed as to resist going in as well as bulging out.

As the author stated, of all the things which should be made standard, a box car side door is one of the most important, and should be the easiest standard to obtain and maintain. Yet, few railroads have cars of different series which have doors interchangeable. I believe in the use of a roller at the top of the door and I am now experimenting with rollers at the top, and with one of the door guides at the bottom equipped with a roller, This will assist in holding the door in place, and I believe will make the door operate much more easily; it will also stop the use of a steel bar for prying the door open or closed. No mention is made in the papers of either a flush or an inside door; both types are being advocated.

There are many designs of steel car lines, and while some answer the purpose for which they were designed, others are unsatisfactory. The tendency is to figure a car line for strength at the center, forgetting the section near the side plate. One function of the car line is to keep the side plate from spreading out as well as coming in, and

therefore, should not be designed to support the roof alone. I had occasion not long ago to inspect a steel car line which had ample depth at the center, but at the ends a part of it was bent at right angles and riveted to the inside of the side plate angle, so as not to furnish any resistance to the side plate going out. I am of the opinion, that if need be, we should sacrifice head room to get car lines nearly straight on the bottom edge, which will act as a tie rod in tension and be in best of shape to withstand compression.

The draft gear and its application to the car are the most important details of any car; yet while this fact is appreciated by all railroad men, a great diversity of opinion exists among them as to what is best. Many are holding to a spring gear, while others claim that the friction gear is a "life saver." Does the good obtained from the use of friction gear warrant the extra expense? There should be a minimum allowable area for draft sills, which should be effective and well balanced about the center line of draft. I have often wished that the standard drawbar height was increased at least 1 in., which would allow a better application of the high capacity draft gears.

The steel frame box car with single sheathing has, I believe, come to stay, with some modification, perhaps, in the general design, to overcome weaknesses which may develop in years of severe service.

C. A. SELEY[1] (written). I acknowledge with many thanks the compliment paid me in the Par. 2 of Mr. Burnett's paper; my connection with the development of composite freight car construction was very interesting and a deeply appreciated opportunity in working out that step in the evolution of car construction. About 15 years ago three factors influenced some progressive railroads to the larger introduction of steel in framework of freight cars: increased capacities, greater structural strength to withstand operating stresses, and the approaching equalization of costs of steel and car lumber, particularly for framing.

For new cars, I believe there is now no good argument against steel for the complete framing, so combined that the sides will assist in carrying the load. The question then arises as to how far to go with the use of steel for such parts of the car as merely contain or shelter the load. Manifestly, floors must continue to be made of wood to enable blocking of the lading. Aside from this there are many predictions of all-steel box cars. In my opinion this will be

[1]President, American Flexible Bolt Co., Chicago, Ill.

the ultimate construction, but general adoption will doubtless be slow on account of the still favorable balance in favor of the cost of wood for lining and sheathing, and in combination with steel plate for roofing, whether of the so-called outside or inside type. When the all-steel box car does come, it will have to be arranged with ventilation features to prevent damage to lading from sweating and from accumulation of excessive heat which may unfavorably affect many high-grade commodities if shut up in a steel box without such ventilation.

The Canadian Pacific Railway is to be congratulated as being the pioneer in developing the inside sheathed box car and, judging from the record as stated by Mr. Burnett, and giving due weight to the statement that "practically no reports of loss or damage to lading due to shrinking" are received, that construction would seem to be fully justified when proper lumber and care in building are used.

Both authors have discussed the advisability of the standard car. I doubt very much if this idea will ever be consummated, even to the extent of the standard material idea advanced by Mr. Burnett. The difficulty in the way is the human element; if we all thought alike it would be possible. The Master Car Builders' Association has standardized the parts essential to interchange, and under this head may be listed couplers, air hose, wheels, axles, journal boxes and contained parts, brake shoes and brake gear parts. The Government has standardized safety appliances. This to the uninitiated would seem to settle most of the difficulties in car repairs, but we all know that very few of the Master Car Builders' standards are really standards in exact detail, and the Interstate Commerce Commission safety appliances necessarily give considerable range of dimensions and application within which their requirements may be fulfilled.

It is difficult for one not in railroad service to appreciate the whole problem, and particularly the influence of interchange requirements. A railroad may be of low gradient, equipped with light power, and have a class of traffic that would ordinarily keep their cars on their own line, and the cars which would most economically fulfill all requirements for such a line and service can be readily imagined. In interchange, however, these cars might be required to go anywhere from coast to coast, in all kinds of tonnage trains, through hump yard trials and other tribulations never experienced on the parent road.

Furthermore, a railroad car designer can never afford to worship standards in view of the rapid evolution in transportation. This may be illustrated by the progress of the Master Car Builders' Association

in adopting standards for car axles. The 3¾ in. by 7 in. axle for 40,000 lb. capacity car was standardized in 1873; the 4¼ in. by 8 in. axle for 60,000 lb. capacity car followed in 1889, a lapse of 16 years; then in only seven years, the 5 in. by 9 in. axle for 80,000 lb. capacity car was standardized in 1896; and that was followed in 1899 by the 5½ in. by 10 in. axle for 100,000 lb. capacity cars, and the end is not yet. The only limit is the gage and endurance of rails for wheel loads, and all above the rail must be proportioned to the stresses.

O. C. Cromwell[1] (written). Referring to Mr. Burnett's design of steel upper frame box car I observe that no diagonal braces are used in the end framing of the car, such as are generally used in the end of a wooden frame car. These braces tend to keep the end framing of the car square, and while their omission would probably not be apparent for the early life of the car, would it not be expected, as the car ages, to find a loosening up of the riveted joints uniting the posts with the plates and underframe? While the car is new, the end sheathing will serve to keep the framing straight, but will not these end boards loosen up in time through shrinkage?

The paper states (Par. 12) that the lumber in these cars is very carefully selected, only specially dried lumber being used. When the car is new, this, of course, tends to keep the framing square, but as this class of car becomes more numerous, there will be a letting up in this feature of obtaining specially dried lumber, and the shrinkage of the boards will be more pronounced, and their effect as a brace will become less efficient. Also we know that on gondola cars the side planks decay under the side stakes and under the corner bands. Would not a similar action of the lumber be expected in this character of car as time went on? Would not this, in turn, lead up to the loosening of the framing? Also will there not be a tendency for water to get into the planks through the bolt holes and start decay?

The irregular distribution of the loading in cars throws twists into the superstructure, and this has a tendency to rack loose the riveted joints. Also, the jacking up of cars to remove the wheels from the trucks throws similar strains into the framing.

Provision has been made in this car for taking up the shrinkage in the sheathing by providing slotted bolt holes in the framing for securing the bolts holding the sheathing in place. By slacking up these bolts and straining down the tie rods, the boards are to be

[1]Mechanical Engineer, Baltimore & Ohio R. R. Co., Baltimore, Md.

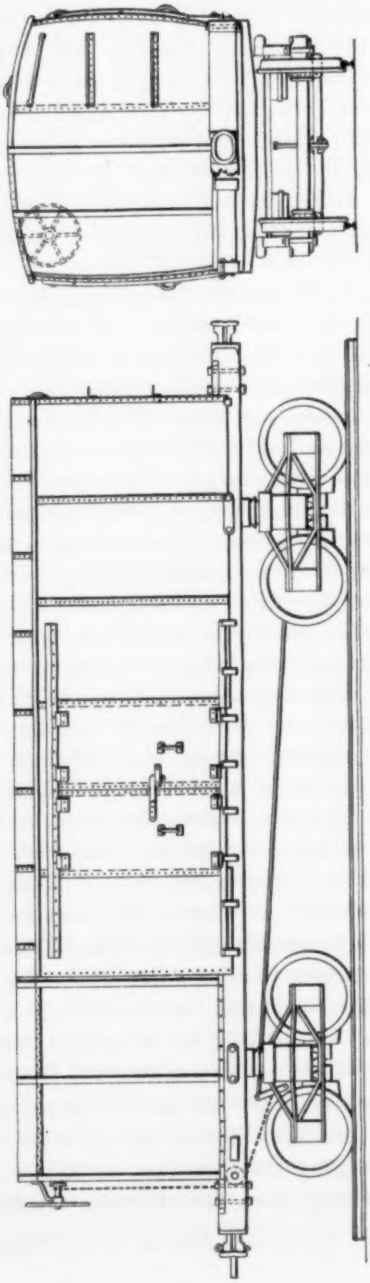


FIG. 15 DESIGN OF IRON BOX CAR BUILT BY THE BALTIMORE & OHIO R. R. IN 1862

drawn to tight joints. Recently I had an opportunity to look over the outside frame box cars of another line, and found that they were not using the slotted bolt holes above referred to, and upon inquiry, was informed that these boards could not always be drawn tight by the use of tie rods, and that since they would not shift into place by the use of the tie rods, their use could well be dispensed with.

I would like to inquire if there is any provision made to forestall the tendency of sheathing to decay back of the posts and braces. At one time, the Baltimore and Ohio operated some outside wooden frame box cars with inside sheathing running longitudinally, and considerable trouble was experienced with the sheathing deteriorating or rotting back of the posts and braces. There was also considerable trouble in maintaining the superstructure of these cars in a square condition, and a different form of roof bracing construction had to be resorted to. I would be interested to learn if the outside frame cars are going to develop a little later on the same peculiarity as to rotting of the sheathing that developed in the old wooden cars.

I understand that the joints in the boards are tongued and grooved, and I have been informed that on some of them the tongues and grooves are made a little deeper than is usual in order to insure against opening to the weather in the event of the boards shrinking. In case the boards shrink, being in a horizontal direction, has there been any trouble, during driving rains, with the water being driven around the ends of the boards and working into the inside of the car? I would like to inquire if anyone has had any experience with other forms of board joints than the tongue and groove.

Recently we have seen the introduction of steel ends to wooden frame box cars, and it occurred to me in connection with the end framing for steel outside frame box cars, that notwithstanding the fact that the 1¾ in. thick boards were used in the ends, these boards will be broken out by the shifting of the loads, and the application of the steel plates to the ends of the cars would protect the boards, and act as a stiffener to the end of the car. Possibly this phase of the question may have been given some consideration and it is mentioned at this time in order to bring out this feature.

Has there been observed any tendency for the sheathing boards to swell, due to the changes in weather conditions? I believe that it has been the general experience that when kiln-dried sheathing is applied to box cars, and driven up too tightly, trouble will result in damp weather with the boards swelling, and the sides of the car buckling.

This is experienced with narrow widths of sheathing applied vertically. With sheathing of wider widths of boards laid horizontally, in which the moisture is retained longer, would it not be expected that there be more swelling of the lumber, and for this reason, it may be that the slotted holes in the posts for securing the sheathing are useful more for the purpose of taking care of the swelling and shrinking of the lumber than for taking up any wear.

I believe that there are some outside frame cars in which the siding is applied vertically; I trust that there will be some points brought out in regard to this paper that will give some light on the advantages and disadvantages of the two methods of applying the sheathing.

Referring to the corrugated steel lining used instead of lumber for sheathing, the Baltimore and Ohio Railroad in 1862 built some iron box cars, which had wooden underframes, but the body and roof of iron plates. Fig. 15 shows one of these cars that had a body 24 ft. ¼ in. long, 8 ft. 2 in. wide, and about 6 ft. 6 in. high. The side and end plates were $^3/_{16}$-in. sheets, applied vertically, the sheets being about 37 in. wide, and the roof sheets about 32 in. wide, all sheets being riveted together at the joints. The sides were slightly convexed to give them stiffness, as was also the roof, but the end plates were applied perfectly straight. The sides, ends and roof were stiffened with 2-in. by 2-in. ribs of ash. These cars proved unsatisfactory, because in summer they became so excessively heated that they spoiled the merchandise, and in sudden changes of weather, produced sweating, with damage to lading, and the cars had to be finally withdrawn from service and used for special trade, and were ultimately converted into workmen's storage sheds, tool houses, etc. I would be interested to know if any of the corrugated iron cars have shown up any of these defects.

Referring to Mr. Rink's paper, the several tables bring out the principal dimensions in comparison very clearly, and in looking them over, one is impressed that in the main, the dimensions are very close, and that it would need only the guiding influence of some association to eliminate these variations to bring them within very much narrower limits. The metal parts, rolled shapes, etc., used in the construction of these cars, could be found in the stock of any railroad, in case the cars were damaged or needed repairs.

The side posts and end posts, side braces and end braces, corner posts and door posts, should be brought down to a standard; they are now very nearly this, as shown in the tables. The side posts, Table

4, are of the same dimensions for all the cars, except in the case of the 4-in. pressed steel shape, while there is only a slight variation in the side braces, and they should be easily reconciled. Several different cars have the same corner posts, and as they are very much exposed members, it is important that they be alike on all cars. Table 6 shows that the end posts are very close, and have very little variation.

The point brought out in Mr. Rink's paper, with reference to the height of the floor above the rail is certainly an important one, and it appears to me that there is no good reason why we should have a variation of $6\frac{3}{4}$ in. in this height, as brought out in Table 7; this, too, affects largely the height of the truck. As it is desired to work towards standard and interchangeable truck parts, the height of the truck is an important one to bear in mind.

W. S. Atwood (written). The first steel frame box cars built by the Canadian Car & Foundry Company were considerably heavier than those now being built. The car referred to in Fig. 1 of Mr Burnett's paper weighed 41,200 lb.; this, of course, was partly due to the fact that the car was constructed from stock material, no special material having been ordered. In fact, the side braces and posts were made of heavy angles in place of the light Z-bars, as at present in use and as originally intended. There are several details which have been changed from time to time in order further to increase the efficiency of the car. These details have also had a tendency to lighten the weight, so that now a car weighs less than 37,000 lb.

Referring to the grading of the lumber, with the first cars of this type, there was some difficulty experienced in securing the proper grade of lumber and also in drying it properly for application to this type of car, as owing to the thickness, the lumber had to remain in the dry kilns longer than was necessary in the case of the thinner sheathing used on the outside sheathed car, and on this account the car companies were not, in all cases, equipped with sufficient dry kiln capacity. The lumber dealers, however, have met the car builders in attempting to prepare a satisfactory grade of lumber for these cars and no difficulty whatever is encountered at the present time in getting an adequate supply of a satisfactory grade.

As to the allowance for finishing, in the first lot of cars built by the Canadian Car & Foundry Company, owing to the condition of

the lumber it was necessary to use, it was thought advisable to purchase 6-in. stuff to finish 5-in. face, but owing to the amount of this grade of lumber which has been used in the last two or three years, it is now possible to get a satisfactory grade, purchasing 6-in. stuff to finish $5\frac{1}{4}$-in. face, which makes possible quite a saving on a large order of cars.

Judging from the service which this type of car has given to date, and also from the favorable criticism it has received from practical railway men, it would seem that it would eventually replace the outside sheathed car, and owing to the simplicity of the design it would seem possible for the railways to adopt a standard car of this type in so far as the general dimensions and sections are concerned. The adoption of a car with standard inside dimensions and outside clearances would be a matter of considerable importance to the car builders, as material could be stocked and available for building cars required for quick delivery for any of the railways which had adopted this type of car. It would also allow them to place large orders for the inside sheathing which could be prepared and stored a sufficient length of time to make it entirely suitable for this type of car and eliminate any possibility of trouble from shrinkage.

From Mr. Burnett's experience with one of these cars lined with corrugated steel, and also from the experience which others have had with steel lined box cars, it would seem only a question of time until the wood inside sheathing would be replaced with steel. I also believe that, in addition to the lining eventually being made of steel on these cars, some type of all steel roof will also be adopted, thus eliminating wood entirely, with the possible exception of the flooring.

I do not agree entirely with Mr. Rink in his statement that engineers did not understand the importance of low fiber stresses in the early designs of steel underframes. There is no question about this not receiving the same attention in the early designs as it is receiving at the present time, but I think that the desirability of showing as favorable a cost and weight of the all-steel and steel underframe cars as against the all wood cars had a great deal of influence on the early designers. Since these early designs have gone into service a great deal of information has been accumulated in regard to these stresses and the importance of strong construction is now receiving more consideration, so that there is not the same comparison made between the steel car and the wooden car as was made by the early advocates of steel car construction.

The statement of Mr. Rink, that vertical sheathing is preferable to horizontal on account of its protecting the lading against water, is hardly borne out by the results which have been obtained with cars with the horizontal sheathing. With cars constructed with a suitable grade of lumber and with reasonable care at the joints, there should be no possibility of the lading becoming damaged with water. In fact, upon inquiry from officials who have several thousand of these cars in service not one complaint was known to have been received where the water had damaged the lading. I think the horizontal sheathing has many advantages over the vertical.

The advantages that Mr. Rink shows would result from all the roads having cars built of this type using uniform sizes of material cannot be emphasized too much. If any large number of railways using this type of car could agree on a uniform inside dimension and a uniform size of sheathing, it would certainly facilitate the procuring of material and would result in eventual benefit to the railways. It would enable the car builders to arrange with the lumber manufacturers for large supplies of this particular lumber, which could be thoroughly dried and satisfactory for use, thus eliminating any need of rushing the material through the kilns in cases of short delivery.

On the first cars of this type, the diagonal braces in the end panels, which Mr. Rink refers to, were applied, but owing to the stiffness given to the frame from the heavy inside sheathing used, both the side brace and the end braces were omitted after the first few cars were built. There has been no trouble experienced from insufficient support for the corners of the car and by using the end braces as verticals the advantages against protection from shifting loads would warrant, I think, their being used in this way instead of as braces. The corner of this type of car at the eaves gets a great deal of support from both the side and end lining and judging from the service which cars have given under my observation, it would seem that this support was ample.

I would heartily favor Mr. Rink's recommendation as to the use of door hangers with rollers and to convince one of the desirability of this feature one would only have to put in a little time at the freight sheds to see the difficulty which is experienced in opening a large number of side doors.

As regards the standard height from the rail to the top of the floor, I think there is no dimension in the construction of the car which it is more important to have uniform than this, in order to

facilitate as much as possible the unloading of box cars at platforms and also to facilitate their being used as runways when placed between tracks from which freight is to be transferred.

F. M. Whyte (written). Both papers touch upon the subject of car roofs directly and, in addition, refer incidentally to conditions which affect roof design and maintenance. The steel underframe is getting to be pretty well understood and is quite universally used, and information about the steel upper frame is accumulating rapidly, so that it is fair to presume that the remaining part of the box car, the roof, will be given more careful consideration in the near future.

In Mr. Rink's paper, mention is made of possible loose rods, posts and braces, having a serious effect upon the roofs by permitting too much weaving of the upper structure, while in Mr. Burnett's paper it is explained that some twisting of the car body is desirable to permit the car to adjust itself to uneven track; apparently some weaving of the body is desirable and yet it is possible to have too much, which is believed to be the general opinion of those who have given consideration to the subject. However, independent of this concededly necessary weaving of the body of the car as a whole, the end framing needs, from the standpoint of the general framing and from that of the roof, more consideration than has been given to it. The end plate is a very important member of the body framing; it ties together the ends of the side plates; it receives a large proportion of the shocks given to the end posts by the shifting of the loading; and it receives the momentum of running boards, brackets and ridge pole when the car is suddenly started or stopped. Further than this, in many of the designs of end framing shown in Mr. Rink's paper, the outer ends of the end plate act as cantilevers in assisting to transfer the load on the ends of the side sills to the center sills through the end posts.

The end plate is also an important part of the roof proper because in all types of roofs some member of the roof is attached to the end plates. As more attention is given to the roofs in the future it is probable that something will be done to the side and end-framing which will assist the roof. There are a few railroad men who think that the roof, meaning that part which sheds water and carries the roof loads, can be safely used to assist in keeping the rectangular box of the car body in proper shape, but the great majority think that this is quite impossible, at least within reasonable weights for the roof. It seems much better, from an operating standpoint and from

the standpoint of weights, to permit the body to weave more or less and to provide a flexible roof to cover it.

There is no very good reason for the confusion which appears to exist concerning what constitutes the roof; aside from the car lines, the roof proper is the material used to support the roof loads and to shed water. With the inside roof, the outside roof, and the roof with two courses of boards and a course of felt between them, the roof boards are quite as essential as are the steel plates in the first two, and the felt in the last one, and when the roof boards are damaged the *roof* is damaged. Mr. Rink says that the inside roofing sheets should last for a good many years if the roof boards are replaced every four years; this means that the life of the roof as a whole is four years. The roof boards cost about as much as the roofing sheets and too frequently this fact is overlooked.

Concerning the weights of roofs, the double board roof with felt between weighs the most; as between the inside and outside roofs with the same thickness of steel, the inside will generally weigh somewhat more than the outside roof on account of the additional framing necessary to support the roof boards above the metal sheets of the inside roof. However, the weight per square foot throughout the car, is practically the same for both; the all steel made of No. 16, U. S. gage will weigh least. An all steel roof of plates $1/8$ in. thick, referred to by Mr. Rink, would weigh more than either the inside or outside roofs with No. 22 U. S. gage steel roofing sheets, but the thickest sheets of which the writer knows is $^3/_{32}$ in., which would weigh somewhat less per sq. ft. than the inside or outside roofs even though the latter have roofing sheets of No. 26 U. S. gage. The No. 24 U. S. gage sheets are generally used for either the inside or the outside roof and as it is the mean of the two extremes of No. 22 and No. 26, referred to by Mr. Rink, it will be used in the following comparative weights.

Considering roughly that the area of the roof is 400 sq. ft. and neglecting additional thicknesses in joints, covering strips, etc., the weights per roof would be about as follows: double board roof, 2640 lb.; inside or outside roof No. 24, U. S. gage sheet, 1728 lb.; all steel No. 16 U. S. gage, 1020 lb.; all steel $^3/_{32}$-in. thick, 1530 lb.; and all steel $1/8$-in thick 2040 lb. Thus the all-steel roof of No. 16 U. S. gage steel will weigh about 700 lb. less than the inside or outside roofs with No. 24 U. S. gage roofing sheets.

There is also considerable misunderstanding about the first cost

of various types of roofs, whereas under present conditions there is not much difference in the costs of the various types. With the double board roof the extra course of boards and the felt will cost about the same as the metal sheets of the inside and outside types. The all steel types with No. 16 U. S. gage galvanized steel or $^3/_{32}$-in. black steel will cost little if any, more than the other types. However, considering the ultimate cost to the railroad company, it is necessary to go somewhat deeper than the first cost of the roof; for instance, Mr. Vaughan has said in the discussion of these papers, that it costs $12 per ton per year for hauling, but some prominent railroads put this cost as high as $22 per ton per year; however, if the mean of these, namely $17, is accepted it will be understood how important it is to reduce to a minimum, the material used to turn water and to carry the weight of a man or two. The cost per year for hauling 1000 lb., whether the rate per ton per year be $12 or $17, will more than pay the interest on the total cost of any of the present types of roofs including the car lines; this fact ought to impress those who are interested in ultimate costs.

Another item in connection with the roof which affects to a considerable extent the total cost of the car is the total thickness of the roof including the roofing material and the car lines and the manner in which the car lines are attached to the side plates. There is no good reason why the distance from the top of the floor to the top of the side plates should not be the inside clearance height of the car; nevertheless, in many instances the under face of the car lines is placed 3 in. to 4 in. below the top of the side plates and in such designs, the side and end framing and sheathing are the same 3 in. to 4 in. longer than necessary and entirely needless weight is added thereby, frequently to the amount of 300 lb. to 400 lb.

George W. Rink. The discussion has brought out some interesting facts, and in some cases a difference of opinion still exists as to the advisability of a standard box car. While it is true, as mentioned by Mr. Seley, that the capacity of freight cars had materially changed since 1873, we have reached a stage of development wherein it would be of immense benefit to all concerned if one standard type of car were to be purchased by all railroads, this design to remain standard until changes are required to permit the reinforcement of some detail which experience has shown must be modified. A number of railroads are now having cars built from designs practically the same as those developed by them as far back as

four or five years ago, which give excellent results in service. It is true that changes must be made from time to time, but they can be made along standard lines by a representative committee of railroad men assigned for this purpose. Suppose it was decided, after building a standard car for several years, that a radical change in design was necessary, some benefit would certainly be derived in having all cars purchased in that period built along the same standard lines.

My object in tabulating the sizes of various members of the box car, particularly the steel upper frame car, was to call attention to the small variations in weight and size of material which exist at the present day; and as mentioned by Mr. Kiesel, this material is difficult to obtain on short notice, and more attention should be given such details, with a view of using only standard materials which it will pay the manufacturers to keep on hand.

H. H. Vaughan, in closing the discussion for Mr. Burnett, said that he felt the author was correct in that it was not a question of how long it took to get structural material, but that it was a question of not having to get material at all. It was very rare that structural material was required for repairs on the steel superstructure car. If the car was side swiped, which was one of the worst forms of damage, it was only a question of cutting the members apart, straightening them out and riveting them together again. The amount of material required in repairs of steel cars of any type was exceedingly small compared with the material required for repairing wooden cars, whether hoppers, box cars, or any other type. The only parts taken out were those that had been crowded together, or torn apart, and it was rare to find pieces torn apart under wrecks. As a rule they were in such shape that they could be cut apart and replaced.

As to the introduction of a standard car, Mr. Chenowith advocated a standard car, and then proceeded to describe a new type of car which he thought was an improvement over those in use. In the cars which Mr. Burnett called standard there was one change after another; little improvements were suggested as each lot of cars was built. The Canadian Pacific began building cars a couple of months ago, and improved them in the making by changing all the rivet spacings. It was found that the rivet spaces were inconvenient in many cases and they could be built much more easily by changing the rivet spaces. If a standard car were decided on tomorrow, the next order for cars

would contain some improvements and changes. It seemed to Mr. Vaughan that the best plan was to follow standard parts and to use material easily obtainable. With steel construction, that question would largely take care of itself.

In answer to Mr. Cromwell, Mr. Vaughan said that a number of different joints were tested before the tongued and grooved joint was decided upon. This style of joint, also a joint with a steel piece inserted between the rabbet and lining running into a steel strip were experimented with. The tongued and grooved joint was found to be the best, however, in the experience of the Canadian Pacific Railway.

In reply to an inquiry as to whether or not the features of car were patented, Mr. Vaughan said that that was rather a difficult question: there were some patents on this type of car, but so far as he knew no one had ever paid any royalty on them, and he did not know of any important patents that covered the construction of this type of car.

R. W. Burnett. I will make only a few remarks on some items not replied to by Mr. Vaughan or not fully covered in the paper.

Mr. Kiesel questions the advisability of using single sheathing on account of damage due to driving nails for securing blocking, etc. It is my observation that there is nothing to fear from this source, as even our oldest cars of this type show no signs of causing trouble. This is also true concerning the absence of diagonal braces in end framing. These cars even when severely stressed, in accident, retain their squareness in end frame to a surprising degree.

If I understand Mr. Chenowith correctly, he considers that relieving the side frame of load stresses will render the car more flexible. I do not believe this would make any appreciable difference in the flexibility of the car. In his remarks concerning strength of end plates, I presume he has reference to that type of car using vertical sheathing, as horizontal sheathing assists the end plate very considerably in resisting the reaction of the end posts.

Mr. Cromwell's remarks on braces and end framing are answered in my reply to Mr. Chenowith, except that I may say it is needless to worry about twisting strains due to jacking car, as the twisting stresses received in a few miles' journey over track that has heaved, produce more racking than all the jacking the car would receive in a lifetime.

No. 1406

# COTTON CONVEYING SYSTEMS: THEIR SAFEGUARDS AGAINST FIRE

By H. A. Burnham, Waltham, Mass.

Member of the Society

The purpose of this paper is to describe briefly methods of limiting the number and extent of fires such as occur from known causes in cotton mills in connection with the preparing processes of opening and picking. Such fires form from 50 to 75 per cent of all the cotton mill fires reported. The particular point in these processes to which attention is directed is the mechanical arrangement of the conveying apparatus used for transferring the cotton from the bale opening room to the picker room.

2 The methods described are the results of the experience of many of the best known cotton mills in the northern and southern states, whose coöperation with the Associated Factory Mutual Fire Insurance Companies has been instrumental in developing this class of apparatus along the present lines of greatest safety and efficiency. There are records of many instances in which, by the observance of the simple precautions described, troubles from fire formerly occurring with great frequency and regularity have almost entirely disappeared.

## MACHINERY

3 The usual machinery for the above work consists of:

*a* A bale breaker or opening machine having a spiked apron moving at moderate speed at the rear of a square iron hopper, which is fed by hand or by a mixing apron with lumps of cotton from the bale. This machine loosens up the matted cotton, allowing any heavy foreign substance to drop out, and delivers the loose cotton uniformly to the conveyor. The general appearance of this machine is shown in Figs. 1, 2 and 3.

*b* The conveyor, either pneumatic or mechanical, or both, carrying the loosened cotton to the picker room. Fig. 2

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

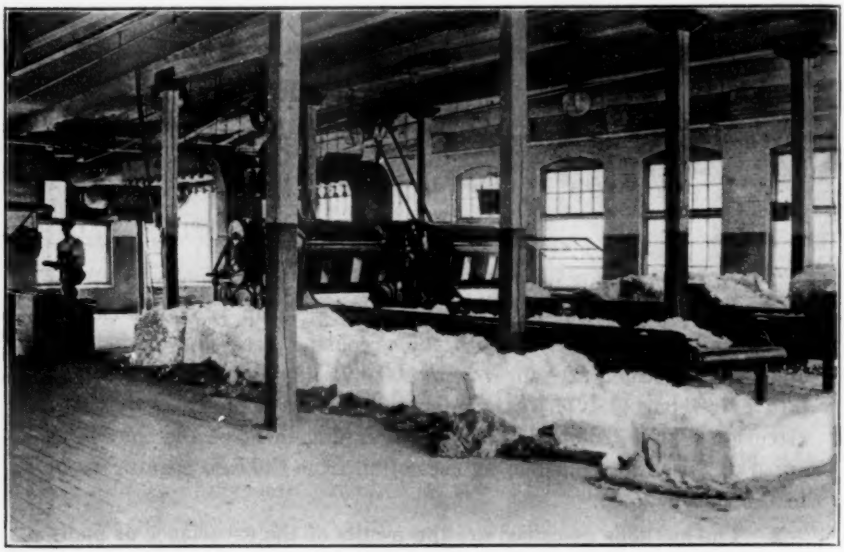


FIG. 1 OPENING ROOM HANDLING LARGE QUANTITIES OF COTTON FED BY HAND FROM THE BALES TO THE APRONS

FIG. 2 OPENING ROOM WITH BALE BREAKER DELIVERING TO A CONVEYOR PIPE UNDER SUCTION, WITH BOX TO CATCH HEAVY MATERIAL

FIG. 3 PICKER ROOM WITH HOPPERS OF AUTOMATIC FEEDERS FILLED BY HAND FROM PILES FED FROM CONDENSERS ABOVE

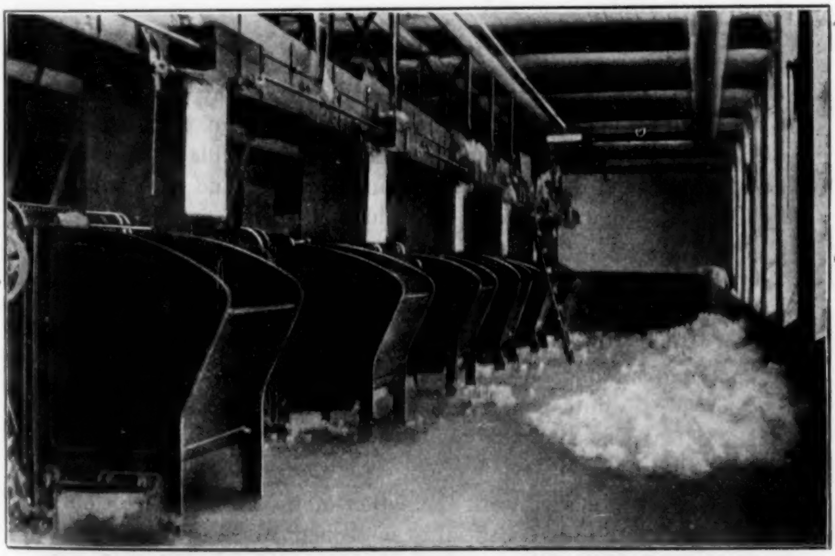


FIG. 4 PICKER ROOM WITH HOPPERS OF AUTOMATIC FEEDERS FILLED BY OVERHEAD MODERN MECHANICAL CONVEYOR

shows the feeding-in end of a pneumatic conveyor, and Fig. 4 shows a mechanical conveyor.

*c* The pickers which have hoppers and feeding rolls to deliver the fiber to iron casings containing rapidly revolving metal beaters. The latter, with the assistance of air currents, separate the fiber from the dust. A line of these hoppers being fed by a mechanical conveyor, is shown in Fig. 4, and another line in Fig. 3, without any feeding mechanism.

4 As fires are almost sure to occur if heavy foreign material or matches enter the pickers or the fan of a pneumatic conveyor, it is important that all such material be removed at the bale breaker or before it is reached and that the various parts of the conveyor be properly arranged.

5 None of this foreign material is more difficult to remove than matches, as they weigh too little to separate readily from the cotton by ordinary means. In this matter more than in any other, preventing the matches from entering the stock is the only true solution of the problem. This has been actually accomplished in some mill properties by providing the help with safety matches to the exclusion of all other kinds.

### PNEUMATIC CONVEYORS

6 The pneumatic cotton conveyor as it exists today is the outcome of attempts made in earlier days to save the labor of trucking baled cotton from the storehouse to the picker room. Mills having storehouses separate from their manufacturing rooms, and especially the larger mills, found this labor a considerable item.

7 At that time it had been common practice to convey damp cotton from dye house to drier by blowing it through a long sheet metal pipe attached to the discharge outlet of a pressure blower, the stock being fed in through the suction inlet of the blower (see *a*, Fig. 5). In applying this method to dry cotton fed by hand, however, fires in the blow pipe were common, and the air blast made them so intense that the arrangement in this crude form was considered far too risky for general use.

8 In many kinds of work the action of the air on the cotton in transit was found to be beneficial to subsequent working of the cotton; so much so, indeed, that manufacturers who had not at first thought it worth while, wished to use the blowing system.

9 The causes of these fires were found to be foreign substances in the stock, such as bits of iron, stones, or matches which had been carelessly dropped, or the wedging of the cotton in the fan casing due to the feeding in of masses which were too large to pass through. In some cases this fault was aggravated by the too small clearance in the fan between the blades and casing.

10 The number of fires with this arrangement was considerably reduced in some cases by using blowers having bronze blades. The introduction of the bale breaker or opening machine above mentioned

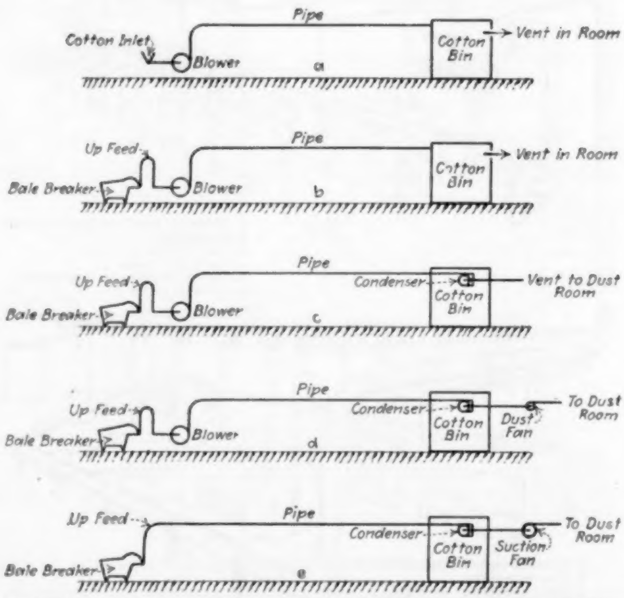



FIG. 5 DIAGRAM SHOWING DEVELOPMENT OF SUCTION SYSTEM OF PNEUMATIC COTTON CONVEYOR

in place of the irregular hand feed also contributed materially toward the elimination of fires from these causes (see *b*, Fig. 5).

11 The general introduction of the cotton condenser in its improved form, however, furnished means for avoiding the objectionable blast of air and cotton in the bin or room, and of operating the entire system under suction instead of under pressure. This advantage of operating under suction entirely, however, was not at once recognized, and many systems were installed with the stock blowing through the fan discharging to the condenser (see *c*, Fig. 5), or with an auxiliary

fan beyond the condenser smaller in size, and known as a dust fan (see *d*, Fig. 5).

12 The condenser consists of a slowly revolving cylindrical screen in a casing, with air pipe connections so arranged that the air current in passing through it, deposits cotton on the outer surface of the screen from which it is removed by a small roll, the cotton dropping lightly from this roll.

13 The final step to the present practice was to eliminate the blow

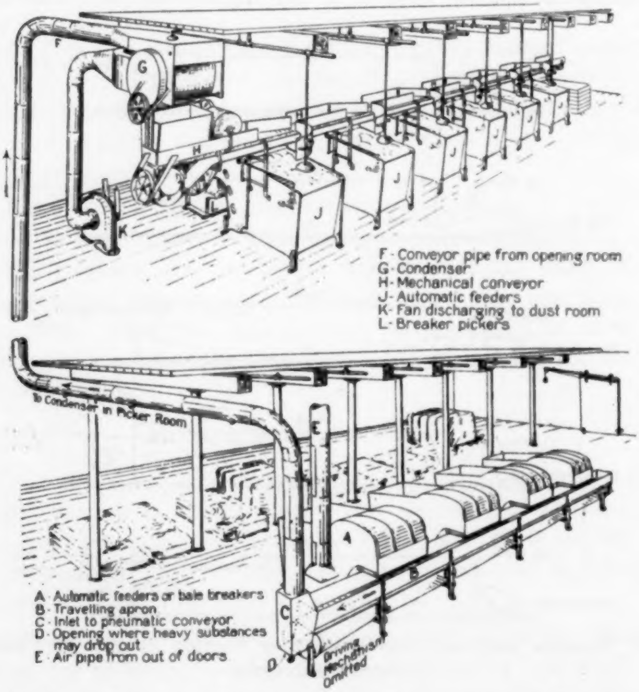



FIG. 6 SUCTION PNEUMATIC SYSTEM TAKING COTTON FROM AN OPENING ROOM AND DELIVERING TO A MECHANICAL CONVEYOR FOR DISTRIBUTION

fan, and to make the dust fan large enough to produce the necessary air current, thus making the cotton-carrying part of the system entirely under suction (see *e*, Fig. 5).

14 By placing the condenser and the cotton inlet both on the suction side of the fan, as in Fig. 5*e*, or both on the discharge side of the fan, as in Fig. 10, no stock passes through the fan, and with systems so installed fires are a rare occurrence. For convenience in

reference, the former is termed "Suction System" and the latter "Pressure System."

### SUCTION SYSTEMS

15 Under the most favorable conditions, pneumatic systems arranged under suction have been successfully handled through pipes up to 900 or 1000 ft. in length, quantities up to 4000 lb. of cotton per hour. The limiting distances and weights of cotton are determined by local conditions such as size and tightness of pipes, number of bends, uniformity of feed, and adjustment of air passages through the condenser.

### ESSENTIAL FEATURES OF A MODERN CONVEYING SYSTEM

16 The sketch, Fig. 6, shows a modern safe cotton-conveying system taking cotton from an opening room through a suction pneumatic conveyor and delivering through a condenser upon a

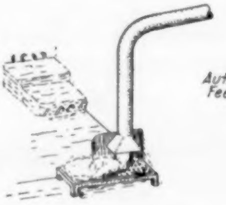


Fig. 7 Arrangement in Opening Room with Bale Breaker Omitted

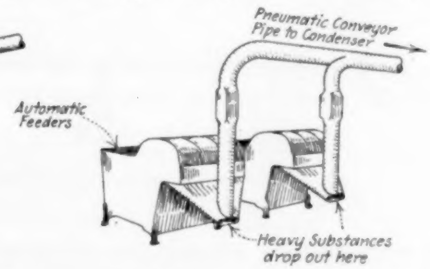



Fig. 8 Arrangement in Opening Room with Mechanical Conveyor Omitted

mechanical conveyor by which the cotton is carried to various points. This combination represents the best modern practice in the larger mills. In mills handling various staples, the discharge is often into bins instead of into machines.

17 In planning a new system for cotton conveying and distributing, or in remodelling an old system, the following conditions in the relative arrangement of parts should be approached as closely as possible in order to eliminate unnecessary fire hazards:

18 *Removal of Foreign Substances.* Provision should be made for heavy substances to drop out of the stock before entering the conveyor pipe. This can be done by having the cotton pass vertically upward into the conveyor pipe, shaping the inlet like an inverted funnel or box over the feed table or apron, and making the vertical

part of the pipe about 12 ft. high (see Fig. 6). To obtain the desired vertical height it is sometimes necessary to use a long radius inverted U.

19 *Location of Fan.* The fan should be so located that no cotton will pass through it. In cases of extreme length this condition can be met by installing two separate systems in series with a relay station at which the condenser of the first system discharges to the inlet of the second.

20 *Disposal of Air Vents.* Air currents from the system should not discharge into the main rooms. This condition is usually met by discharging the air direct to the dust room.

21 *Location of Pipes.* Long pipes should be generally located outside of main rooms and should not pass through important fire walls or floors. Trouble from condensation inside the conveyor pipe in cold weather, which sometimes arises, may be avoided by providing an auxiliary cold air inlet (see *E*, Fig. 6), and in difficult cases by

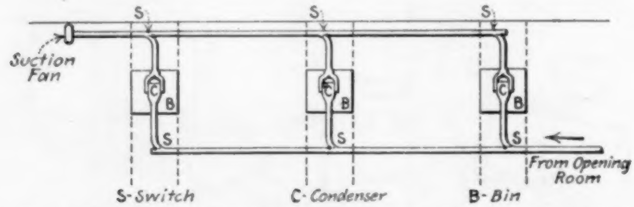



Fig. 9 Plan Showing Arrangement in Picker Room with Mechanical Conveyor Omitted

covering the conveyor pipe, where out of doors, with non-conducting covering.

22 *Pipes* should have joints riveted in addition to being soldered and have suitable handholes for cleaning. This is to guard against breakage of the pipe in case of fire from inside or outside the pipe.

## VARIATIONS IN ARRANGEMENT

23 Variations from the arrangement shown in Fig. 6 are frequently necessary, depending on the number of varieties of stock, the quantity, quality, and distance to be carried, as follows:

24 *Omission of Bale Breaker.* In the opening room of the smaller mills, the bale breaker or automatic feeder is often omitted and the cotton fed from a pile previously shaken out by hand from the bale. In this case the stock should be fed across a slatted movable table under

the inverted cone of the inlet pipe (see Fig. 7). Uniform hand feed, however, is difficult and without the automatic feeder, frequent clogging of the condenser may result.

25 *Omission of Mechanical Conveyor in Opening Room.* In some of the older systems or where more than one pneumatic conveyor is to be fed, the mechanical conveyor in the opening room is sometimes omitted by forming a pipe connection between the automatic feeders and the branches to the conveyor pipe. In such cases this connection may be in the form of a flattened conical pipe sloping downward and having an opening at the lowest point opposite the upward turn where heavy material may drop out (see Fig. 8).

26 *Omission of Mechanical Conveyor in Picker Room.* At the

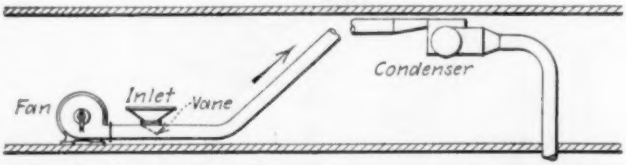



FIG. 10 ARRANGEMENT OF INJECTOR INLET FOR PRESSURE SYSTEM

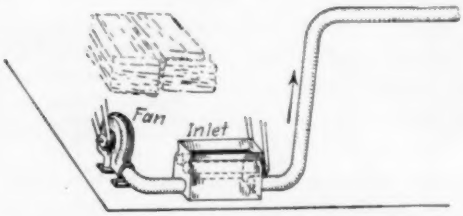



FIG. 11 ARRANGEMENT OF FEED BOX INLET FOR PRESSURE SYSTEM

picker room end, in case several staples are used, the mechanical conveyor is often omitted and the cotton delivered direct to bins through condensers. Under these conditions the condensers may be piped in parallel with proper switches and the same fan used for all (see Fig. 9).

## PRESSURE SYSTEMS

27 In handling cotton mixed with some wool, as in knitting mills, felt mills, or colored cotton mills using a large variety of colored stock, pneumatic systems are usually operated under pressure instead of suction. The reasons for this are, that (*a*) stock containing above a small percentage of wool cannot be successfully handled with condensers, the greasy nature of the wool causing it to stick and clog the

mechanism; and (*b*) in handling colored cotton the bins are often so small and numerous that the use of a condenser for each is impracticable. Mechanical conveyors are also objected to by manufacturers for handling colors on account of the liability that small tufts of stock caught in the mechanism will afterward find their way into lots of another color.

28 The pressure system improperly installed is dangerous to life and property, as the strong blast of air intensifies any fire in the pipe or near its outlet. The safest arrangement of such a system is obtained by observing the conditions already stated regarding location and construction of pipes for suction systems, and also the following conditions:

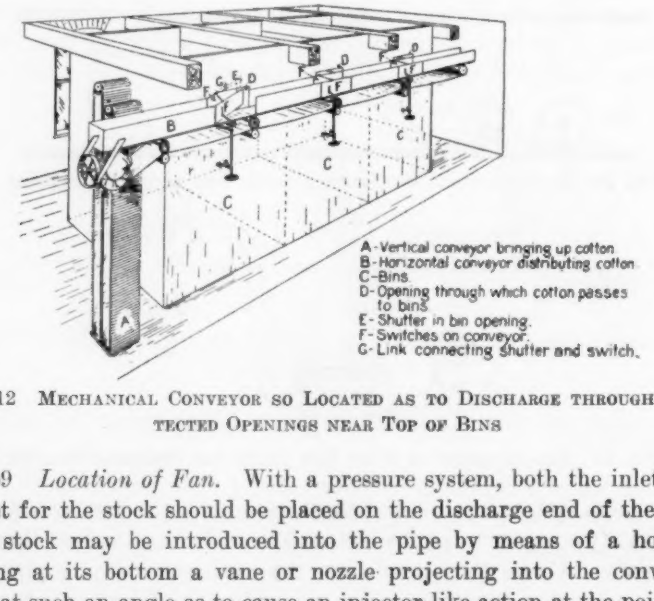



FIG. 12 MECHANICAL CONVEYOR SO LOCATED AS TO DISCHARGE THROUGH PROTECTED OPENINGS NEAR TOP OF BINS

29 *Location of Fan.* With a pressure system, both the inlet and outlet for the stock should be placed on the discharge end of the fan. The stock may be introduced into the pipe by means of a hopper having at its bottom a vane or nozzle projecting into the conveyor pipe at such an angle as to cause an injector-like action at the point of inlet (see Fig. 10). This device is practicable, however, only for distances up to about 75 ft. Another method adapted for longer distances is to use a pair of flexible surfaced rolls forming the upper part of an air-tight feed box (see Fig. 11).

30 *Vents.* At the delivery end, with either of the above arrangements, the pipe should terminate in a bin, with tight walls, with vent through a screen piped to such a point that no blast of free air can enter any important room.

MECHANICAL CONVEYORS

31 As with the best of machinery and with all the safeguards thus far described, all fires cannot be eliminated from the picker room. it is important that the extent of such fires as may still occur be made as small as possible. This is especially desirable in mills which store temporarily in bins in the room considerable quantities of opened cotton of various long staples.

32 Mechanical conveyors are often used between the opening room and picker room bins. In such cases these rooms are usually near each other. The older method of passing these conveyors through openings in partitions between adjoining bins has proved objectionable, as fire may enter several bins at the same time. To avoid this objection the conveyor can be placed over the tops of the bins, discharging through protected traps in the top. This arrangement requires considerable head room.

33 Another arrangement is to place the conveyor high up at the front or rear wall of the bin and discharge it through small protected openings in the wall. This can best be done where the bins are not deep from front to back and where the conveyor has discharge from its side (see Fig. 12). In either case the openings for feeding bins from mechanical conveyors should be protected by tin covered shutters interlocking with the switch of the conveyor so each shutter will be open only when its bin is being filled.

SPRINKLERS

34 Having limited the number and extent of fires, the matter of controlling promptly those which do occur, should be cared for. In addition to the automatic sprinkler equipment always needed at the ceiling of the cotton working rooms, sprinklers should be placed in the bins, in the condensers, under traveling aprons, inside and under picker trunks and close over the cotton in the hoppers of the automatic feeders.

## DISCUSSION

Albert W. Thompson. Referring to the objection of Mr. Burnham to the using of relay fans, through which the cotton can pass directly, in cases of extreme length, in place of making substantially two systems, that the cotton in passing through the fan is likely to strike a fire due to the clogging, possibly from a poorly installed fan,

from hot-bearings at the fan, or foreign substance passing through, if they get by the first point where they are supposed to drop out, do I understand this statement is based on actual experience? I have seen one or two systems of considerable length arranged in that way, and I do not recall now ever having seen a fire traceable directly to the fan. Most fires have originated in some other place, either in the opening room, where a device something like a picker is sometimes used for tearing bleached or dyed cotton, and where matches sometimes start fires; or at the finisher picker room at the other end of the system; but I do not recall having seen a fire directly traceable to the use of a relay fan.

C. J. H. Woodbury. I believe this paper is the first of its type which has been presented before any of the technical societies. That these occurrences in and of themselves are not preventable in any given instance appears to be conceded by Mr. Burnham, who correctly represents the practical experience in the matter. Many years ago, phosphor-bronze picker blades were introduced because they would not strike fire, or so it was stated; but whether that allegation was proved in practical experience beyond a diminution of such fires, is extremely doubtful. There is one class of distribution of such fires where it has been prevented, and that is the fires caused by striking in the picker and spreading down into the dust-room below, and then backing up through the dust flue into another picker which was not then in use. That has been prevented by unbalanced dampers at the end of the dust flues, so that when a picker was not in use, those dampers would naturally close and prevent that. I would like to ask the speaker if the principle of the automatic link, which has been applied so successfully to fire doors has not also been applied to dampers in these flues, so that whenever a fire occurs a flue would be closed at a very early stage in the story?

The illustrations represent the method of feeling the opener or bale breaker directly over open bales of cotton; whereas in many mills they put an ageing pile for the purpose of letting the cotton receive the air, and also get the effect of time in resuming its spirility, and its contact with other fibers, which it has lost by the compression it has undergone in the bale. There are various types of these ageing stacks in the English mills. They use them for the purpose of blending the various grades of cotton, or of mixing with the cotton the card waste, to the extent of making a new art almost in the blending of cotton for specific purposes, of which the American cotton manu-

facturers, I regret to say, know practically nothing. The cotton is put in strata in the pile from the various types of cotton, the Egyptian cotton, the American upland, and not forgetting as large a portion of card waste as the fabric will stand. Then they rake down the cotton along the end of the pile after it has remained there a week or so, with a large wooden garden rake, and a blending is effected in that manner. Another question is: Whether the exhaust blower is as efficient in moving cotton through rarefication of the air as a pressure blower would be?

John A. Stevens asked what data the author had on fires caused by matches, referring to the paper in which fires started by matches are mentioned; also where the large number of matches came from.

E. V. French. Mr. Woodbury has touched upon the matter of ageing cotton. Where ageing is done large mills may open 50 or 100 bales of cotton at a time. If a fire occurs under such conditions it is almost sure to run over the whole pile in spite of any extinguishing apparatus and cause a good deal of loss. Many mills on coarse work have found where the cotton is handled by blowing, that the passage through the pipes in loose form, and with an ample air supply, accomplishes about the same result as the ordinary ageing. Only 10 or 12 bales therefore need be opened at once and the cotton from these is carried directly to the automatic feeder at the inlet of the blowing system. This gets rid of large amounts of loose open cotton, and makes a distinct gain in the reduction of the fire hazard. Picker rooms are causing a large number of our cotton mill losses and improvement at this point is the next thing to take up. If we can eliminate a good many of these fires the cotton mill will become about as safe as the average risk. In the question of matches which Mr. Stevens mentions, I remember one case where arrangements were made through a public officer to search the men in a cotton mill; the result was two handfuls of matches found in their pockets. As they work a match will now and then drop from their clothes to the floor, to be stepped upon or run over by a truck, and strike fire which will surely ignite anything combustible nearby. This is undoubtedly one of the most serious causes of fires in such mills today.

Charles H. Bigelow, in answer to Mr. Stevens, said that in the South he had seen the negro workmen handling matches and a num-

ber of fires caused by dropping matches, which were cracked by the trucks passing over them and set fire to the cotton, sometimes catching in the cotton on the truck.

THE AUTHOR. In reply to the questions of Mr. Thompson, I know of one system where two fires occurred. It was a particularly long system, the fires blowing through the outlet side of the system, but no trace of fire had been found on the suction side of the fan.

Replying to Mr. Woodbury, in regard to stopping fires passing through the dust flues from the pickers by means of the fusible link, I do not think that method is extensively applied. I have heard of places where they tried to apply that method to cotton passing through pipes but I think the fire usually gets by the fusible link, as it takes a considerable length of time to melt the solder on the link. As to the unbalanced dampers, they are very efficient and I have no objection to urge against them. They are a great protection against fire backing up in the pipes.

The reference to the ageing qualities and the blower system, has suggested to my mind the fact that the quality of the cotton is improved by being blown through the pipes. I know of a number of cases where ageing bins have been put into mills for trial and then discarded as the passing of the cotton through the pipes was found to answer almost the same purpose. That may not be true of all cases, but it is in many. As regards the relative efficiencies of suction and blower fans, I have no data on that point; I think very few accurate measurements have been made to determine this.

Referring to the question of Mr. Stevens, the data are not exact, for as a rule in such cases the matches are destroyed. Sometimes we find the remains of the match, and often we find hard substances that have passed through. But many fires ascribed to matches are no doubt caused by matches, and we reach that conclusion from the fact that every other cause of fire we know anything of has been eliminated. That, perhaps, is not absolutely conclusive. The match fires, however, still remain the hardest ones to contend with or to trace to a cause.

In the South I think the laborers are largely responsible for many of the fires due to matches. The matches may be dropped by workmen around the gins, or even at earlier stages, before they get to the gins. It is a common practice among the negroes of the South to stick matches in button-holes, in pockets, back of their ears, and in their hat-bands.

## No. 1407

# SPECIFICATIONS FOR FACTORY TIMBERS

By F. J. Hoxie, Phenix, R. I.

Member of the Society

This paper is given in the hope that a discussion may be opened which will bring together scattered experiences relative to the properties underlying the qualities required in mill timber. Within three years more than $100,000 has been required to repair damage due to dry rot in mills insured by the Associated Factory Mutual Fire Insurance Companies, and nearly all of the lumber affected was hard pine in comparatively new buildings. This rapidly increasing destruction of lumber is unquestionably due to the use of poorer material than was employed in former years. In a lot of large timber for the frame of a new mill will be found nowadays scarcely anything but North Carolina, Cuban, or Shortleaf pine. The best qualities of these varieties are probably as resistant to decay as is the Longleaf pine, but they are less uniform; the poor qualities are much more difficult to detect because of the necessarily superficial acceptance inspection of the purchaser or his representative.

2 In textile mills the high artificial atmospheric humidity undoubtedly increases the tendency to rot in pine of low natural resistance to fungus. The weaving rooms of cotton mills are frequently maintained at a saturation of moisture of 70 to 80 per cent. With 70 per cent saturation and a temperature of 80 deg. fahr. a decrease in temperature of 12 deg. or more would cause precipitation. This condition can be found in winter on the under side of roofs or at beam wall bearings or windows if the heat insulation is not very complete.

3 Dry rot fungus will grow on the surface of wood at atmospheric saturation from 96 to 100 per cent[1]; it will grow inside of large beams of susceptible material at a much lower atmospheric saturation. In one case investigated, the room was maintained at 60 per cent and the beams of the second floor which were not exposed to outside temperatures were attacked. Experiments tending to show that the dry rot fungus is able to form sufficient water for its requirements in

[1]Hausschwammforschungen, vol. 6, p. 308; Moeller, Jena, 1912.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

FIG. 1 LONGLEAF PINE, 8 IN. BY 18 IN. DISTRIBUTION OF RESIN: *A* 25 PER CENT, *B* 14 PER CENT, *C* 17 PER CENT, *D* (SAPWOOD) 3 PER CENT

beams of large size by decomposition of the wood are given by Mez[1] Wet spots frequently remain several days on the freshly sawed surfaces of beam sections containing living fungus (see Fig. 3). There is no question whatever about its growing inside of factory beams of non-resistant material for two or three years at least after the factory is built, with less than 70 per cent of atmospheric moisture and in rooms where water is not used in manufacturing processes. This condition has been found in several of the worst cases investigated.

4 Only hard pine timber will be considered here as this is the most generally used material for heavy mill frames. Spruce is sometimes used for floor plank, but this is probably more susceptible to rot than the better qualities of hard pine. In the only case investigated where spruce was concerned the disease evidently spread from pine beams to the spruce floor planking. In a few cases hemlock without antiseptic treatment used under floors in basements has failed in a few years. Hemlock, oak and white pine have failed after longer or shorter service, but the use of these materials for mill construction is not sufficiently frequent to require more than passing mention.

5 Twenty-six varieties of hard pine, growing in North America, are described by Penhallow;[2] of these only four are much used for timber in the eastern states: the Longleaf (*Pinus Palustris*), Shortleaf (*Pinus Echinata*), Cuban (*Pinus Heterophyila*), and North Carolina (*Pinus Taeda*). They are very fully described in Timber Pines of the Southern States, by Charles Mohr.[3] The Longleaf pine is the most valuable for lumber from its greater strength and durability. The latter quality is probably due chiefly to the high percentage and uniform distribution of the resin and to the small percentage of sap wood, which averages not over 2 in. in radius of mature trees, while that of the other three varieties varies from 4 in. to 6 in. of the radius.

6 Many of the specifications for hard pine are vague or meaningless owing to the fact that the terms used are not clearly defined or are not determinable. For example "best quality" could honestly be taken to mean the best in the local market or the best in the world. "Longleaf pine" is practically impossible to identify beyond a doubt in the form of lumber, and in fact it is now rather the exception than

[1]Der Hausschwamm, Carl Mez; Dresden, 1908, p. 192.

[2]A Manual of the North American Gymnosperms, D. P. Penhallow, 1907.

[3]Bulletin No. 13, U. S. Department of Agriculture, Division of Forestry, 1897.

FIG. 2 CUBAN PINE, 8 IN. BY 18 IN. WITH ROT IN SAPWOOD

the rule to find this variety of pine in the large sizes required for mill beams. Other varieties are regularly accepted as Longleaf. "Georgia" is sometimes added to a specification but lumber grown in that state is seldom insisted upon, nor is its place of growth important.

7 The Interstate rules for Yellow pine lumber for 1905 define prime quality, dimension sizes, as follows:

All square lumber shall show two-thirds heart on two sides, and not less than one-half heart on two other sides. Other sizes shall show two-thirds heart on face and show heart two-thirds of length on edges, excepting when the width exceeds the thickness by 3 in. or over, then it shall show heart on the edge for one-half the length.

The rules define merchantable inspection as follows:

All sizes under 9 in. shall show some heart the entire length on one side. Sizes 9 in. and over shall show some heart the entire length on two opposite sides, etc.

This specification for prime quality loses most of its force when applied to pine of the type shown in Fig. 4. The young heart wood has rotted nearly, or quite as rapidly, as the sapwood. Most of the beams shown in Fig. 6 are of this variety. Timber of merchantable inspection in addition to the uncertainty of the heart wood mentioned above has the practical certainty of rot in the sapwood if conditions are not most favorable. This grading is particularly treacherous with the deep, narrow sections frequently used for double beams.

8 The average number of annual growth rings per inch has been suggested as a criterion for selecting good hard pine lumber, as follows:[1] first quality should have 15 or more rings per inch, as an average of 5 in. across the grain; second quality should have an average of from 8 to 15 growth rings, and material having less than 8 should be considered unfit for structural use. This would admit such material as that shown in Fig. 4 and exclude such as that shown in Fig. 5. The dense coarse grained resinous Cuban heart wood is as strong and durable as the Longleaf pine, its bad quality being that it has a large percentage of not very clearly marked sapwood which is rapidly destroyed by fungus (see Fig. 2).

9 A later suggestion which has been adopted by the American Society for Testing Materials[2] calls all good material Longleaf pine and poor material Shortleaf pine. This is unfortunate in that an arbitrary meaning is given to the names Longleaf and Shortleaf, which already have a generally accepted botanical meaning; such a discrimi-

[1] American Society for Testing Materials, Proceedings 1909, p. 291.
[2] American Society for Testing Materials, Proceedings 1912, p. 345.

Fig. 3 North Carolina, or Loblolly, Pine, 8 in. by 18 in. with Rot in Non-Resinous Wood

nation is also incomplete in not defining the qualities of the two classes.

10 The positive identification of Longleaf pine lumber is difficult or impossible. There are slight microscopic differences in the form

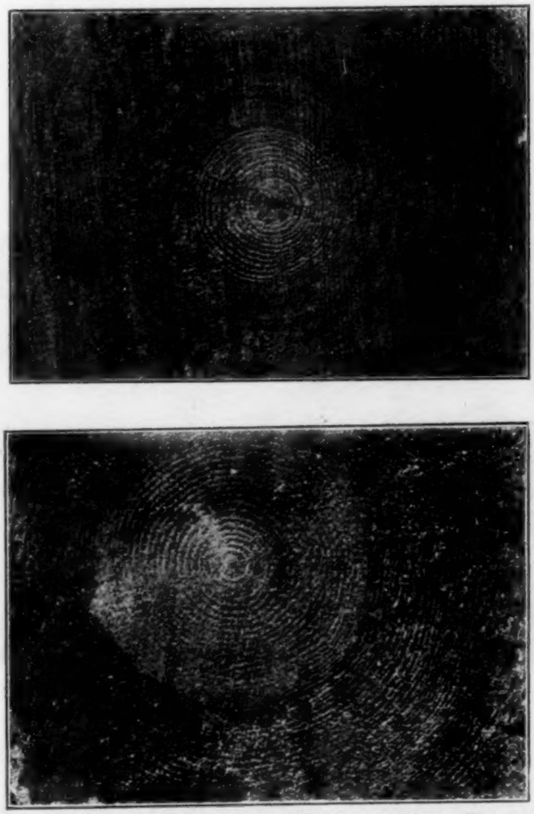


Fig. 4 Sound and Rotted Sections of 12 in. by 14 in. Beams; Probably Shortleaf Pine

of the medullary rays of the North Carolina pine, as given by Penhallow[1] and Roth,[2] but to use them for identification would require considerable experience in the microscopic study of the pines. The macroscopic differences are more striking with characteristic specimens. The Longleaf pine is heavy, resinous and fine grained, averaging for the dry heart wood from the butt of the tree, according to

[1]North American Gymnosperms, D. P. Penhallow, 1907.
[2]Bull. No. 13, U. S. Division of Forestry, 1897, p. 143.

Sargent,[1] 43.7 lb. per cu. ft. (see Fig. 1). The Cuban pine is slightly heavier, averaging 47 lb. per cu. ft., resinous and coarse grained with a large proportion of dense summer wood and a much larger proportion of sapwood (see Figs. 2 and 5). The North Carolina pine is lighter, averaging 34 lb. per cu. ft., somewhat less resinous, coarse grained with a small percentage of summer wood (see Fig. 3). The Shortleaf pine is moderately light, averaging 38 lb. per cu. ft., the least resinous of the four and of medium coarse grain. The following are the weights per cubic foot given by Mohr for kiln dried material; Longleaf 36 lb., Cuban 37 lb., Shortleaf 30 lb., and North Carolina 31 lb. The reason for Mohr's weights being less than those of Sargent

FIG. 5 BEAM 8 IN. BY 12 IN., WEIGHING 72 LB. PER CU. FT.

is that he takes the average of the tree, the top of which is much lighter, while Sargent considers only the denser heart wood at the butt. There is considerable confusion in local names: Mohr gives six Latin and 29 common names as having been used at different times or different places for the Longleaf pine; the other varieties have nearly as many. The Cuban and Longleaf pines are sometimes confused since both have long leaves, while the Shortleaf and the North Carolina have short leaves.

11 A treacherous type of hard pine is shown in Fig. 4, which is probably Shortleaf pine, although it may be Longleaf grown under unfavorable conditions. The percentage of resin and the gravity is

[1] Silva of North America, C. S. Sargent, 1897.

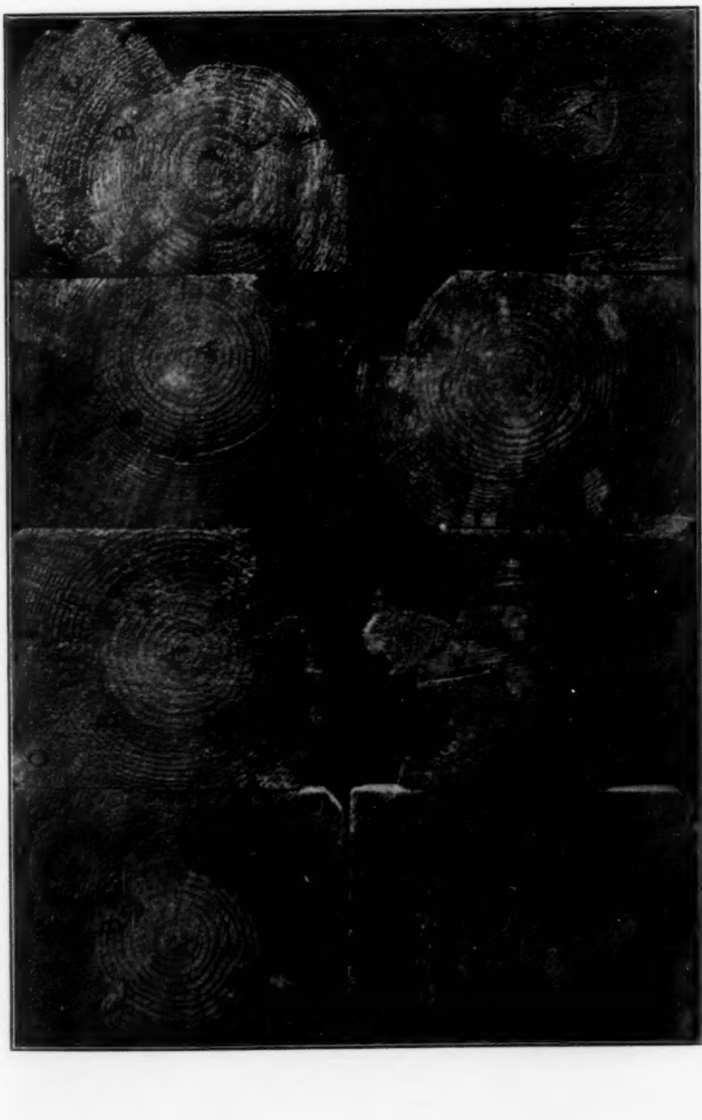


Fig. 6 Rotted 12 in. by 16 in. Pine Beams Removed After Two Years' Service

Resin, Per Cent

| A | B | C | A | B | C | A | B | C | A | B | C |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 7.72 | 5.12 | 0.83 | 8.36 | 4.89 | 1.17 | 4.46 | 1.99 | 0.76 | 5.36 | 7.04 | 0.68 |
| 9.45 | 2.31 | 0.82 | 3.45 | 18.81 | 0.99 | 6.26 | 6.56 | 0.58 | 29.21 | 12.17 | 0.88 |

**Fig. 7 New Selected 12 In. by 16 In. Beams**

Resin, Per Cent

| *A* | *B* | *C* | *A* | *B* | *C* | *A* | *B* | *C* | *A* | *B* | *C* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 14.6 | 2.7 | 1.6 | 12.2 | 2.4 | 2.9 | 34.7 | 14.1 | 2.6 | 13.1 | 14.1 | 2.6 |
| 12.3 | 3.0 | 2.4 | 8.6 | 3.2 | 0.8 | 12.3 | 3.0 | 2.4 | 7.5 | 11.3 | 1.1 |

Air Dry, Lb. per Cubic Feet

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 37 | 42 | 37 | 36 | 39 | 37 | 46 | 57 | 61 | 40 | 44 | 34 |
| 34 | 36 | 30 | 37 | 42 | 33 | 39 | 45 | 36 | 42 | 55 | 50 |

somewhat higher than that given by Mohr[1] for Shortleaf, but the arrangement of growth rings corresponds with his description. Hardly a lot of lumber is to be found of which this does not form a considerable proportion. Most of the rotted beams shown in Fig. 7 are of this type. It is characterized by a coarse grain at the center, generally consisting of four to six growth rings per inch, becoming regularly finer with increased diameter until at 6 in. or 8 in. from the center there are 20 to 30 growth rings per inch. The resin and resistance to rot decrease with the rings. At the center there is generally in the neighborhood of 10 per cent of resin; at 4 or 5 in.

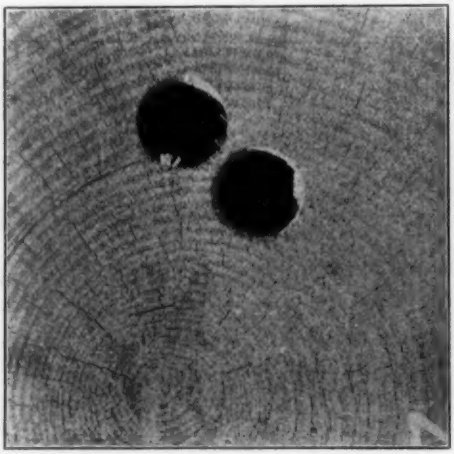


FIG. 8 VIEW SHOWING HOW HOLES IN COLUMNS SERVE AS PASSAGEWAY FOR FUNGUS

from the center there is frequently less than 3 per cent. The line between the heart and sapwood is well marked, the heart being much darker than the sap, as with the Longleaf. This material is regularly sold as Longleaf.

12 Tannin in the lignified heart wood is said to retard fungus growths in oak,[2] but resin is the important factor in the hard pines. The following experiment was made to show the effect of resin: A block of Longleaf pine 2 in. on a side, containing 18 per cent of resin, was sawed in two across the grain. Half of it was boiled in benzole and after the removal of the resin the benzole was driven off.

[1]Bulletin No. 13, U. S. Division of Forestry, p. 13.

[2]Berichte der deutschen botanischen Gesellschaft, vol. 29, 1912, p. 704; Mycologisches Centralblatt, vol. 1, 1912, pp. 138 and 166.

Both pieces were cultivated in contact with wood containing living dry rot fungus. At the end of a year the specimens were dried and weighed. That from which the resin had been removed had lost 8 per cent in weight, the other only 2 per cent.

13 Cuban pine is the heaviest and frequently the most resinous of the timber varieties. Fig. 5 shows a section of a beam weighing 72 lb. per cu. ft. It can be seen that besides being saturated with resin this is nearly all summer wood.

14 The ideal specification consists undoubtedly of a clear description of the qualities required. In structural timber these are strength and durability. With these qualities defined, attributes such as

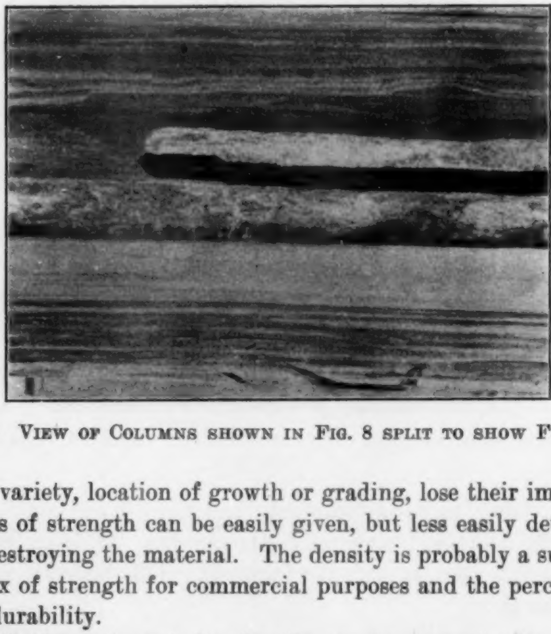


FIG. 9 VIEW OF COLUMNS SHOWN IN FIG. 8 SPLIT TO SHOW FUNGUS

botanical variety, location of growth or grading, lose their importance. The limits of strength can be easily given, but less easily determined, without destroying the material. The density is probably a sufficiently close index of strength for commercial purposes and the percentage of resin of durability.

15 Both the resin and the density can be conveniently found by boring half-inch holes into the end of the beam, collecting the chips, drying and weighing them, extracting the resin with benzole and weighing it. Cuban or North Carolina pine with 10 per cent of resin is apparently as resistant to fungus as Longleaf pine with the same percentage of resin. Heart without resin is not a saving quality.

16 The percentage of resin in the sound centers of rotted beams taken from a mill was determined in order to get an idea of the amount

required to stop fungus growth under ordinary mill conditions (see Fig. 7). In rotted beams of the poorest of hard pine there is generally a sound center which contains more resin than the remainder of the section. Sometimes it is not bounded by the growth rings but is very irregular, the cause being that resin has been irregularly deposited in the section owing to knots or injuries to the tree. The limits of the sound centers are frequently not the same as those of the heart wood.

17 It is apparent that the limiting amount of resin which is just sufficient to stop the fungus is in the neighborhood of 3 per cent. The

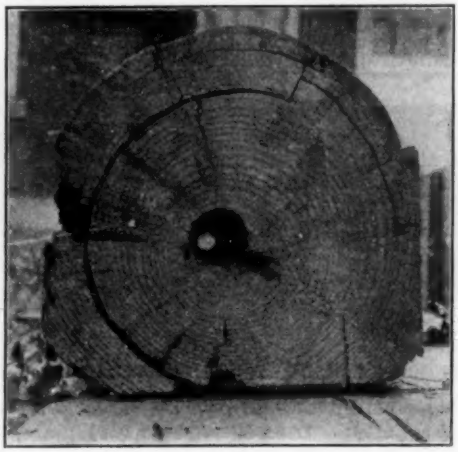


FIG. 10 EXAMPLE OF BORED HARD PINE COLUMN BADLY ROTTED AFTER TWO YEARS

limiting power of resin is undoubtedly not absolute but varies with the moisture, variety of fungus and time of exposure. Therefore, it is safe to assume that a mill beam should have not less than 5 per cent of resin throughout successfully to withstand fungus under ordinary conditions of dampness and allowing a reasonable factor of safety. Dye houses and paper mills would probably require more.

18 In order to determine the distribution of resin in timber of average good quality several sections from beams now being supplied for factory construction were analyzed. Part of these are shown in Fig. 7. The large holes were for samples for resin analysis, and the small ones for density determinations. The material was mostly heart wood and the price paid for it was $59 per 1000 ft. board measure.

None of the sections have 5 per cent of resin throughout. Three would come within this limit by cutting off about 10 per cent and the remaining five would require a reduction of about 75 per cent, as shown by the dotted lines.

19 If it could be logically assumed that better material could be bought by increasing the price in proportion to the reduction of the cross-section necessary to obtain the required 5 per cent of resin, \$115 per 1000 ft. would be indicated. From such figures as I have been able to obtain this price is not far out of the way for such selected material in large sizes, and even at that price it is doubtful if a sufficient amount for a factory could be had in the required sizes. This indicates that hard pine with sufficient natural resistance to withstand

FIG. 11 VIEW SHOWING FUNGUS BETWEEN DOUBLE BEAMS, FRONT BEAM HAVING BEEN REMOVED

fungus unassisted is out of the question at prices that compare favorably with other forms of construction.

20 Low power of resistance to fungus in the timber is frequently the last cause to which a serious dry rot outbreak is charged. Ventilation is generally the first preventative measure suggested. Dry wood which is kept dry is undoubtedly incapable of fungus infection but ventilation does not necessarily cause drying. Wood will become dryer or wetter in proportion to the relative humidity of the air; therefore, timber ventilated with moist air will have its rate of rotting accelerated if it is of a susceptible variety. Moreover the susceptible varieties absorb moisture more rapidly than those which are more resistant to fungus.

21 A heavy coat of paint may accelerate or retard the rate of rotting, depending upon whether it prevents wood from absorbing or

giving up moisture. The question of primary importance is whether the wood is resistant or susceptible to fungus.

22 Holes through columns and narrow openings between beams encourage fungus more frequently than they prevent it. One of the reasons given for bored columns and double beams has been that they prevent dry rot, and considerable importance has sometimes been given to this feature.[1] Undoubtedly columns with holes through them and thinner beams would dry out more quickly, if given a chance to season, but the common custom of boring green or wet columns just before they are put in place in the building and using moist lumber for double beams leave ideal places for the growth of fungus, since the air in the openings is saturated with moisture. Holes in the columns are also objectionable since they form a convenient passageway for the fungus to pass rapidly from floor to floor before the building has dried out. Fig. 9 shows a section of a bored pine column split in two with fungus growing through the hole; the column was not deeply attacked as can be seen from the photograph of the cross-section, Fig. 8. The hole is inefficient in preventing rot if the material of the column is susceptible, as can be seen from Fig. 10, which shows a section of a bored column badly rotted. Fig. 11 shows fungus growing in the opening between double beams, the front half of the beam having been removed.

23 In many old mills hard pine timbers are found to be sound after years of service in moist air. A careful examination will generally reveal the fact that these timbers are resinous Longleaf pine, practically no longer obtainable. There is no more reason for supposing that the non-resinous hard pines now in common use would prove as resistant, than there is for expecting moist hemlock or birch to resist fungus. There is still available for present requirements in mill construction sufficient hard pine of suitable size and strength. It remains to supplement its lower natural resistance with proper antiseptic treatment.

## DISCUSSION

E. M. Bates. I would like to ask Mr. Hoxie how far the action of preservatives prevents this fungus growth inside of the beams? Does the treating of the pine timber with one of these preservatives until it is thoroughly saturated on the outside lead to conditions where dry rot can start on the inside? In other words, is the life

[1]Engineering News, Vol. 62, 1909, p. 620.

of that timber dependent upon the wearing out of this waterproof surface on the outside? Also has he reached any conclusion as to the most effective method for treatment?

A. W. Thompson. I would like to ask what would be the probable effect of carbolineum on material in an average mill, with the ordinary humidity, where artificial heat is used? Also what is the best practice as to painting timber? It has been the general impression that it is best to let them remain unpainted for a year or two after they are put into the mill.

A. F. Ernst. Must these preservatives be applied under a vacuum in order to secure their efficiency? Should timber put into the ground, such as fence posts, be coated wet, or is it absolutely necessary to have it dry before putting on the preservatives?

Charles H. Bigelow. The author speaks of a number of mills having to be reconstructed. How extensive is this danger? I did not realize that the danger was so serious, especially in new work. It is certainly surprising to have timbers rot away in a couple of years and is something that must be looked out for.

Another thing to be considered is the large number of knots in some of the timber. In looking at a building recently where Longleaf pine had been specified and the contractor had furnished something, which he said was just as good, but which most certainly was not, I noticed several joists that had two and three knots one above the other near the center on the bottom half. A timber of that sort can support only a very small load, for there are no continuous fibers on the outside to be stressed and give a good modulus of rupture and unless there are good joists on each side a bad failure might easily occur. In Salem, Mass., where I used to live, is an old house that is over 100 years old where I have noted a case of apparent dry rot, and I would like to know if this is similar to what Mr. Hoxie has been describing. In this case the outside surface of the timbers, which I think are oak, is furrowed with longitudinal tunnels filled with a dry dust. These may have been made by some insect, but I imagined it was dry rot. It has destroyed perhaps a third of the timber but the balance of the wood appeared to be sound and all right and as the timbers were much larger than are used now there appeared to be plenty of strength for years to come. The rot did not appear to be getting much worse the last time I saw it and I have known of it for over 20 years.

E. V. French. I want to emphasize what is really the revolutionary character of the information which Mr. Hoxie has brought to us. He shows that practically all of the old requirements as to timbers cannot be used safely today so that new specifications must be made in keeping with the new facts. Most of the old slow growth timber with its large percentage of resin is exhausted. It will be several generations before timber of this kind will grow again, so that for many years inferior timber which is susceptible to dry rot will have to be used. To make such timber safe Mr. Hoxie's data show that it must be treated with some preservative to avoid rapid decay. It is, of course, feasible in new work to use timber which has been treated, but this is only a part of the problem, for with the increase of humidity in old mills, due to the modern manufacturing requirements, timbers which have remained sound for years have decayed rapidly. Recently in a mill 75 years old, white pine beams, which must have dried out thoroughly years ago, were found to be seriously impaired by dry rot. The cause was evidently the material increase in the humidity maintained in these rooms in recent years. This condition may be very serious from the fire standpoint. We have laid great emphasis on the slow burning character of plank and timber construction, but a beam with its center decayed may have but a thin shell of sound wood which will be weakened after a short fire and allow large sections of floor to fall. We do not wish to raise fears, but we must recognize that in these directions there are new conditions of vital importance affecting all such construction.

The Author. In reply to Mr. Bates, sound wood thoroughly treated with a suitable antiseptic material will probably be found immune from fungus with ordinary mill conditions. Dry rot undoubtedly enters the timbers from the outside, but a very common condition is that in which the timber is already deeply infected when treated. The fungus inside the wood is not destroyed by the thin superficial brush coat. If the lumber is heated sufficiently long in a bath of hot material, the heat may destroy living fungus which is beyond the penetration of the compound. As to the most effective method for treatment, no conclusion has yet been reached. Investigation is still under way, and I hope to be able to present something at a later date. The problem of mill timber is quite different from railway ties or posts.

Answering Mr. Thompson, carbolineum has shown good results when applied to telegraph posts and mine timbers. I should, there-

fore, infer that it could be expected to act satisfactorily on mill beams where conditions are not so trying. This material in common with other coal tar compounds has the disadvantage of increasing somewhat the combustibility of the timber and leaving a black, greasy surface difficult to paint. Paint on a dry, uninfected beam would tend to keep it dry and sterile and therefore be an advantage. With a moist, infected beam the opposite would be the case.

In answer to Mr. Ernst, the thorough treatment of timber by the vacuum process is without doubt desirable for fence posts in order to secure the longest life. With wet or already deeply infected material, I imagine the sterilizing and drying effect of a hot bath would increase the efficiency of the treatment and the increased penetration caused by the vacuum would probably secure a greater permanance. Little service could be expected from any antiseptic superficially applied to wet, diseased wood.

Replying to the question of Mr. Bigelow, as to the extent of the danger from dry rot, Mr. Hoxie stated that he had investigated about a dozen cases of dry rot in the past three years; the damage resulting had been found to vary from a few hundred dollars up to $90,000. Referring to the old house in Salem, Mass., mentioned by Mr. Bigelow, Mr. Hoxie stated that the condition mentioned is probably worm-eaten sapwood.

Mr. Hoxie said it was probable that the fungi which Mr. Woodbury mentioned as dangerous in the decayed timber, were those frequently found in timber lying on the ground, not the dry rot which is proving destructive of mill frames. Referring to the use of hot resin under the floor quoted by Mr. Woodbury, Mr. Hoxie stated that the heat might have value in sterilizing the lumber. A space around beam ends in walls can admit moist as well as dry air. If the wall is cold and the air in the room contains a high percentage of moisture, the ventilating space will do more harm than good.

## CONTRIBUTED DISCUSSION

HERMANN VON SCHRENK[1] (Abstracted). The author has reached some rather startling conclusions, which, briefly stated, are as follows:

*a* The percentage of resin may be taken as an index of its power of resistance to rot in any one particular piece of hard pine timber.

[1]Member of firm, von Schrenk and Kammerer, Consulting Timber Engineers, St. Louis, Mo.

*b* The present hard pine timbers obtainable in the market are practically all (as he designates them) "non-resinous."

*c* All of the pine timber now available has a low resistance to decay, and cannot with safety be used in factory buildings without preservative processes.

The conclusions are so at variance with the actual facts in the case that I deem it necessary to call attention to the obvious errors in the paper.

In regard to resin as a factor in preventing decay, in view of the fact that the author's major contention is based on the relation between resin and decay resistance, one would assume that there would be a good basis for such a statement. The only proof which the author presents is a cultural test made with two pieces of pine wood. He also calls attention to the resin content of some decayed beams shown in Fig. 6. To one familiar with the behavior of fungi, the cultural proof submitted is of no value whatever. The reasons fungi grow on one piece and not on another may be due to the moisture content, to the density of the wood fiber, to the presence or absence of various compounds, and to many other factors not referred to by the writer. The decayed beams shown in Fig. 6 are splendid examples of decayed sapwood, and I have no hesitation whatever in saying that the resin content of the particular wood which decayed probably had very little to do with the fact that this sapwood did decay.

Numerous investigations have been made from time to time with relation to resin and fungus activity. Personally, I am of the opinion that the only function which resin has with respect to fungus growth is to act as a waterproofing material. A piece of pine wood having 10 per cent resin would be very resistant to moisture. Resin of itself is not an antiseptic. This may be illustrated by the fact that even in the most resinous trees wood-destroying fungi grow with the greatest ease. I have shown in numerous instances[1] that one of the first steps in the attack of coniferous wood, containing considerable percentages of resin, is the liquefication of the resinous substance, which advances in front of the growing fungi, and in frequent cases drips from old knots. This subject might be discussed at length (note for instance the high decay resistance of comparatively non-resinous woods like cypress and redwood). One quotation, however, will suffice. Tubeuf, in his treatise on Diseases of Plants,[2] says, "Resin is

[1]Diseases of Trees, Year Book of Agriculture, 1900, p. 204, plate 23. See also writings by Tubeuf and Hartig.

[2]Page 62.

in itself not antiseptic," etc. The only so-called antiseptic function which can be ascribed to resin is the physical quality of preventing the entrance of fungus spores and the physical power to prevent the entrance of water. Mr. Hoxie in Par. 12 refers to a possible analogy between tannin and resin. It seems hard to understand why, if tannin is a preventive for fungus attack in oak trees, resin should exercise a similar function in pine trees. The author's citation of volume 29 of the Berichte der deutschen botanischen Gesellschaft is also unfortunate, in view of the fact that no mention whatever of tannin is made in this article. It is unquestionably true that recent experience in various mill construction buildings has shown an increasing percentage of timbers infected with dry rot. I consider it wholly unnecessary, however, to ascribe this to such a problematic factor as the resin content. I have personally examined a good many of the recent cases, and invariably find that where such decay has taken place it has been in timbers having a high percentage of sapwood. Mr. Hoxie shows this in a very striking manner in Fig. 6. It has been recognized for many years that the relative resistance to decay of sapwood and heartwood is very marked. Sapwood decays rapidly and heartwood comparatively slowly. This is of course universally true of timber in which there is a marked distinction between heartwood and sapwood. This is not the place to discuss the reason for this relative resistance. It will be sufficient to point out that it is the *sapwood* factor in timbers which has given rise to recent difficulties, and not the resin factor. The proposal of the author to consider the resin percentage as a standard (Pars. 15-18), and his attempt to figure the value of a piece of timber used on the percentage of resin as a basis, is so extreme, that I hardly consider it worthy of refutation. At best, the engineering profession should go slowly before accepting so radical and unproved a suggestion, based as it is upon a comparatively small number of laboratory experiments, which, so far as can be judged from the paper in hand, failed to take into consideration the numerous conflicting factors usually eliminated in such cultures.

The second point brought out by the author is that the hard pine timbers now available are practically all non-resinous. I have looked through the paper carefully for some statement as to what the author means by "non-resinous" pines. The impression given by the paper is that years ago there were a lot of timbers which come under the group of resinous timbers, but that these have now disappeared.

A possible change which might have taken place in the resinous character of the pines during the last ten or fifteen years, which sug-

gests itself to me, is that during that period of time many of the Longleaf pines have been tapped for turpentine, and this possibly may have given rise to an impression that removing the turpentine from the pine tree makes it less resinous. This is certainly true so far as certain portions of the sapwood are concerned, but it is not true of the heartwood, as evidenced in the report of the Forestry Service Bulletin No. 8.

The resinous materials of the pine tree, which are removed from the tree during the turpentining operations, come from the liquid portions contained in the sapwood resin canals. I am inclined to believe that Mr. Hoxie drew his conclusions from a comparatively small number of sappy pieces; at least I would characterize his generalization as decidedly without foundation. The variable in the resin content of pine timbers is obviously very great, and this is well shown in the analysis which he presents in Fig. 7.

Mr. Hoxie admits that there are still available four species of pine in the South, but he makes the very strong statement that hard pine lumber 12 in. square, or larger, is not obtainable in suitable quantities without antiseptic treatment. It may be of interest to call attention to the quantity of pine timber still standing in the Southern States. According to the last report,[1] of the United States Bureau of Corporations for 1913 there are still standing in the Southern States 384,000,000,000 ft. of yellow pine timber, which the Bureau of Corporations classifies into Longleaf pine and Shortleaf pine. Of the 384,000,000,000 ft., 232,000,000,000 ft. are Longleaf pine and 152,000,000,000 ft. are Shortleaf pine. Referring to the Eastern part of the country, it may be of interest to note that the amount of yellow pine still standing in the Southeastern States is as follows, in billions of board feet:

| | Longleaf | Shortleaf |
|---|---|---|
| Alabama | 25.6 | 12.4 |
| Florida | 58.2 | 0.9 |
| Georgia (part) | 18.5 | 13.2 |
| So. Carolina (part) | 4.6 | 14.6 |
| No. Carolina (part) | 2.9 | 22.7 |

Assuming now that the Longleaf pine available is typical Longleaf pine, no proof has been brought forward by the author that this is less fit for structural use than it was 15 years ago. The resin factor, I believe, can be disregarded entirely. The difficulty of obtaining such Longleaf pine timbers as were available 15 years

[1]Dept. of Commerce and Labor—Bureau of Corporations. The Lumber Industry. Part I, Standing Timber, p. 76, 1913.

ago is, as the writer admits, largely a matter of specifications. In Par. 9 the author criticizes the American Society for Testing Materials' specification for adopting the terms "Longleaf pine" and "Shortleaf pine," going to the extent of characterizing this specification as "arbitrary." I would call attention to the fact that the classification "Longleaf" pine and "Shortleaf" pine is universally adopted not only by scientific authorities,[1] but also by the lumber trade. Reference has just been made to the classification adopted by the United States Bureau of Corporations, in which all Southern lumber pines are classified as Longleaf pine and Shortleaf pine. I would furthermore call attention to the fact that the largest percentage of yellow pine lumber is today being manufactured under the rules of two organizations, the Georgia-Florida Sawmill Association (annual cut about 800,000,000 ft.) and the Yellow Pine Manufacturers' Association (annual cut about 4,500,000,000 ft.) Both of these associations divide all yellow pine into Longleaf and Shortleaf pine (see their standard grading rules).

It was for the reason that these commercial classifications are practically standard that the American Society for Testing Materials indorsed them. The problem of distinguishing between these two grades is a difficult one. I thoroughly concur with Mr. Hoxie that the line of demarcation between the two is very vague, and almost impossible to establish with any degree of accuracy. The American Society for Testing Materials suggests that the distinction between the two be based on the density, that is, on the number of growth rings per inch. This distinction may not be the best, but can be used when liberally interpreted where a botanical distinction is out of the question. Attention is furthermore called to the fact that the distinction on the basis of density, made by the American Society for Testing Materials, was largely from the standpoint of strength. No specific mention is made as to lasting power, because that factor was taken care of in the specification for bridge materials by separate

[1]From the United States Forestry Service, Bulletin No. 108, on Tests of Structural Timbers, 1912, concerning southern yellow pines: "The term 'southern yellow pine' is applied collectively to practically all of the pines of the Southern States which are manufactured into lumber. On the market the manufactured lumber is divided into two classes, Longleaf and Shortleaf. Material with more than 8 or 10 rings per inch, and containing a comparatively large amount of summerwood and less than 30 per cent of sapwood, is called Longleaf pine; while material with fewer than 10 rings per inch, slow-growing material that is light in weight, and which contains much sapwood, is called Shortleaf pine. Commercially, therefore, the terms 'Longleaf' and 'Shortleaf' are descriptive of quality and have little botanical significance."

clauses. Attention is called to the standard specification for structural timber adopted by the American Society for Testing Materials in 1907,[1] in which for No. 1 Longleaf pine stringers the specification calls for timbers which show *not less than 80 per cent heart on each of the four sides,* measured across the sides anywhere in the length of the piece. For caps and sills the specification calls for *85 per cent heart on each of the four sides.* In other words, the American Society for Testing Materials' specification calls attention, first, to the necessity of getting denser timber, and to timber free from sap. Similar specifications are in force in the American Railway Engineering Association. The rule quoted by Mr. Hoxie in Par. 7 as prime, practically says the same thing. It is somewhat surprising, therefore, to read below this: "This specification for prime quality loses most of its force when applied to pine of the type shown in Fig. 4." Reference to Fig. 4 shows the upper timber to be almost wholly sap, and the lower over 50 per cent sap. Certainly neither of these two pieces could by any possible means come within the specification "prime," or the American Society for Testing Materials' specifications for Longleaf pine. The same is true of all of the figures shown on Fig. 6, not one of which would begin to fulfill either of these two specifications.

The chief trouble with a good many of the failures has doubtless been that the purchaser has not given the timber the proper inspection, because with the specifications available, both of the lumber manufacturing organizations and the American Society for Testing Materials, I can see no excuse whatever for anybody putting timbers such as shown on Fig. 6 into a mill of any sort. My main contention is that the writer of the paper has missed the chief point in connection with this discussion, and that is that the important factor in the lasting of Yellow pine in buildings is not the resin content of the timber, but the percentage of sapwood allowed on such timbers. In a recent building failure, which I had to examine personally, in which a large number of timbers had failed, the failure was in timbers with a high percentage of sapwood, and in the practically all-heart timbers there was only one case of decay, and that one was very slight.

If one specifies Longleaf pine timbers for buildings, and strictly adheres to the specification, limiting the amount of sapwood, one will find that the timbers available will give just as good service, from the standpoint of length of life, as those furnished in the past. I agree entirely with the author when he says that timbers of the type

[1]Page 191.

shown on his Figs. 3, 4 and 6 should under no circumstances be used in factory buildings without preservative treatment. I believe, however, that he has given an entirely wrong picture when he accuses all Longleaf pine timber of lack of decay resistance, when such lack is due largely to the amount of sapwood on the timber.

In closing this criticism, I venture to add that many a producer of lumber has been remiss in furnishing sappy timber under a heart specification, and if there is one point which ought to be emphasized with the greatest strictness on the part of engineers, it is that they shall refuse to accept inferior timbers such as those shown in Fig. 6. There has been a marked change in the nature of the timbers manufactured in the Southern Yellow pine field during the past seven or eight years. When Southern Yellow pine was first introduced in Northern markets, it was almost wholly strictly Longleaf Yellow pine. The older buildings were almost without exception built of this heart Longleaf Yellow pine. The early shipments of Longleaf pine usually came from Georgia and Florida. During recent years the pine forests of the more northern states have been drawn upon, and large quantities of timbers are manufactured in North Carolina, South Carolina and Virginia. Most of these are Shortleaf and Loblolly pine, in which the percentage of sapwood is very much higher than in the true Longleaf pine. I have had numerous occasions to protest against the acceptance of sappy Shortleaf timbers (Shortleaf and Loblolly pine) for construction purposes, and frequently it has been difficult to make the manufacturers of this class of lumber understand that they will not be acceptable to the engineer where both strength factors and lasting qualities are desirable. Too much stress cannot be laid upon the necessity for care in not only specifying, but also inspecting every piece of pine timber intended for a building in which both strength and lasting qualities are factors. The poor experience obtained in many recent instances, and specifically those to which Mr. Hoxie refers, are without doubt due to the fact that in these particular buildings sufficient care was not exercised in seeing that these inferior pieces were excluded. I have repeatedly criticized the lumber manufacturers for their failure to live up to their end of th contract. I have also repeatedly criticized the laxity which exists among engineers and architects who allow the contractor or lumber salesman to give them sappy Loblolly or Shortleaf pine where Longleaf pine has been specified. It is highly desirable that the very inspection which the writer of this paper despairs of should be

strengthened, and that wider publicity should be given to the caution to use only heart timbers, and to be sure that you get heart timbers when you buy them.

Summarizing the chief points in this criticism:

*a* Resin cannot be taken as an index of the natural resistance to rot of pine timbers.

*b* There is no basis for the statement that all pine timbers now available are non-resinous.

*c* Longleaf pine timber is still available in large quantities for acceptable factory construction.

*d* Where specifications are written for such factory construction, the greatest stress should be laid upon the percentage of sapwood allowed, because it is the sapwood and the sappy pieces which decay. All recent cases, so far as known to me, have been cases of decay of sapwood timbers, and not of all-heart timbers.

*e* No timber should be put into a building without rigid inspection, to make sure that only such timbers are furnished as are strictly in accord with the specifications.

C. W. Willette. I will briefly give a few facts, based on long years of practical experience which may be of some benefit to others who are engaged in similar construction work.

My field of endeavor includes New York, Pennsylvania, Georgia, Mississippi, Louisiana, Arkansas, Texas, Michigan, Wisconsin, Montana, Oregon and Washington. The timber used includes White pine, Pennsylvania hemlock, oak, elm, Shortleaf pine, Longleaf pine, cypress, red fir, yellow fir, cedar and Western Yellow pine.

I find that one of the most prevalent causes of decay in any of the timbers used, is cutting the tree when the sap is running freely. When the sap is dormant, as in winter time, there is much less danger of decay under like conditions.

It is better to use timber of medium size than to adopt very large timbers on account of dry rot ensuing.

In mill construction I find that timbers more than 12 in. in thickness are subject to dry rot because they are not thoroughly seasoned, especially where the climate is damp.

Another prolific cause of decay is lack of ventilation. Mill sills should be well protected from weather and at the same time be laid in such manner as to be protected from dampness. It is a mistake

to bed sills their entire length on any substance, especially in a damp climate, as moisture is sure to gather and decay will certainly ensue.

Where sills are to be placed upon foundations, a coat of very hot coal tar covering the foundation and another applied to the sill where it is to come in contact with the foundation, will materially prolong the life of the timber.

Where proper precautions are taken under these conditions, any of the timbers mentioned will give very satisfactory results. I know of many buildings which have stood for half a century and the timbers are still in good condition.

White pine carries but little resin and I have had as good results with white pine as with any other timber used in mill construction. Cedar and cypress, oak, and some other classes of timber contain but little resin, hence I do not place much weight on the value of resin alone as a timber preservative.

I find that the following species of timber answer well for general mill work: yellow fir, red fir, Longleaf pine, white pine, Shortleaf pine, Western yellow pine, hemlock, and I would prefer them in the order listed without any radical choice.

In summing up, I will recommend a few simple rules for selecting and placing timbers in mill work, which, if strictly followed, will give good results and lasting satisfaction:

*First:* Be sure that the timbers are cut in the woods while the sap is down, or dormant. This is imperative.

*Second:* So plan the building that it will not be necessary to use any timbers over 12 in. thick, horizontally. If it is absolutely necessary to secure more strength, use two timbers with a space of not less than 2 in. between them. In a very damp climate it is better to use timbers not over 10 in. in thickness, but they may be any size vertically. Well seasoned timber does not decay readily.

*Third:* See that all timbers are well ventilated and that air can get at them freely. This applies especially to lower sills. Where timber is thoroughly seasoned and dried out when placed, and can be kept dry, the circulation of air is not so essential, but it must be borne in mind that dampness is the forerunner of decay, hence dampness to start with must be eliminated so far as possible and dampness must be kept away and the problem is solved. Fungus does not thrive on well dried timber.

*Fourth:* So far as possible, sap should be eliminated from all timber used in mill construction, as it more readily absorbs moisture

than all heart timber, although when sap is kept perfectly dry, it will last as well as heart timber.

I might mention in addition, that in mill construction, I use no mortises or tenons in framing, as I find drift bolts, or dowels better for various reasons. A mortise forms a pocket for moisture.

THE AUTHOR. Laboratory experiments have been by far the least important factors in my conclusions as to the effect of resin in retarding dry-rot fungus. Beams which have rotted in place seriously in several mills under average mill conditions have been used for most of my experiments. Undoubtedly the most important antiseptic function of resin is its waterproofing action. Further observations are without doubt desirable.

Three references will be found in footnotes to Pars. 11 and 12; the last reference shows tannin to be the cause of the immunity in the oak flooring mentioned in the first. This is a discussion of the qualities of pine that rots and of pine that does not rot, not of pine versus cypress or redwood. It is also a question of diseases of mill beams, not diseases of living trees.

Most of the material shown in Figs. 4 and 6 is the conventional Longleaf pine and fills the requirements of Bulletin No. 108 of the U. S. Forest Service, as quoted. I must differ with Dr. von Schrenk as to the sapwood; the percentage is not large and more heartwood than sapwood has rotted. The beam shown in the lower part of Fig. 4 must be Longleaf pine, as it is fine-grained; it must be prime quality as it is nearly all heartwood. Nevertheless it is unquestionably rotted. After a timber has rotted it is very easy for almost anybody to say that it was poor material, but it is frequently much more difficult to predict what it will do before it has rotted.

Dr. von Schrenk states that such timbers as those shown in Figs. 3, 4 and 6 should under no consideration be used without antiseptic treatment. If so, as most of these are conventional Longleaf pine, of prime quality, it is logical to infer that all hard pine timber would require antiseptic treatment to assure its lasting qualities.

I think I have not said that there has been any change in the true Longleaf pine except the use of its name for other varieties. The statistics quoted, giving an estimate of the conventional Longleaf pine standing in the South, do not prove that it will be found more resistant to decay when it is cut into timber than the present lumber supply, which is admittedly less resistant than that of former years.

In the following Dr. von Schrenk agrees with my statements:

*a* Hard-pine mill frames erected within the last seven or eight years show greater tendency to rot than those erected earlier.

*b* The name Longleaf pine now has an arbitrary meaning without botanical significance.

*c* The distinctions between lumber which is conventionally called Longleaf and that which is conventionally called Shortleaf are vague; much is left to the generosity of the lumber dealer and the good judgment of the inspector, and lumber dealers are frequently without generosity and inspectors without judgment, accentuating the need of concise specifications.

*d* True Longleaf pine is as good now as it ever was.

*e* The general run of Southern pine is much poorer now than in former years.

But we disagree as to the value of the percentage of resin as an index of the resistance of hard pine to dry rot in mill lumber; as to whether or not heartwood is a guarantee of fungus immunity; and as to the necessity of general antiseptic treatment of hard pine for mill frames.

I am not trying to demonstrate that resin as a waterproofing agent, or otherwise, retards the progress of rot. This has been a matter of common knowledge for centuries. What I do want to determine is how much resin is necessary to retard dry rot sufficiently in mill timber to make it reasonably safe for use in mill construction with average conditions. I further want to know what the chances are of getting such resistant lumber from the present lumber market. If it is not practicable to be certain of getting such timber by methods in general use, I want to decide upon an antiseptic treatment which can be practically applied and which will render the poorest varieties of timber sufficiently reliable to be used with safety, so that the mill owner may be reasonably assured that his expensive structure will not be seriously damaged by dry rot in a few years and that a rotted column or beam will not cause an accident which may kill his employees. Any assistance which Dr. von Schrenk can offer along this line of endeavor will undoubtedly be appreciated by the mill engineers of the country.

## No. 1408

# TEXTILE COST ACCOUNTING

## A BRIEF TREATISE ON ITS PURPOSE AND APPLICATION

By C. B. Annett,[1] New York

and

C. F. Cunningham,[1] New York

Non-Members

Cost accounting, as the name implies, is the application of the science of accounting to the determination of costs. Its fundamental principles are applicable to all lines of industry. It is really only a detail of the large question of organization, and it is therefore evident that this detail cannot be solved independently. There must be:

*a* "A clear definition of the departments of the business so that similar work and similar functions shall be grouped in one department.

*b* "A positive definition of duties of each department and its relation to the general business.

*c* "A clear fixing of the responsibility and a willingness to have responsibility rest with the heads of the various departments.

*d* "Scrupulous care to communicate the various wishes of the Chief Executive through the proper channels, that the responsibilities fixed may be preserved and each department limited to its particular functions.

*e* "The harmonizing of the commercial accounts, that is, the representative accounts of the business with the general scheme which is to be followed in the factory accounting or costkeeping department. . . .

*f* "A proper head to the department of costkeeping who must be as much an engineer as an accountant, and capable not merely of compiling figures, but of using the information when the facts are compiled; for the end of costkeeping is cost reduction. This man must be so efficient that he

[1]Gunn, Richards & Co.

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

> may be depended upon by the highest official of the company and he will naturally be high in the counsels of the latter. . . . A man who fills such a position will have no sinecure."[1]

2 It is perfectly obvious that costs which are inaccurate, are not worthy of consideration and in fact are worse than none at all, as they are often entirely misleading. It therefore seems desirable before entering upon a discussion of specific methods of obtaining costs, to discuss briefly the fundamentals essential to all cost accounting.

3 The first essential is that cost should be a statement of fact, all assumptions, in so far as possible, being eliminated. In contradistinction, an estimate, or estimated cost, may not agree with the real cost because of the very fact that it takes into consideration probable changes of conditions.

4 The lack of a clear understanding of the difference between a cost and an estimate, has frequently led to much confusion and the distinction between the two cannot be too clearly defined. It seems to be a weakness of many manufacturers to fall into the habit of allowing themselves to believe that cost is what they wish it to be, rather than a statement of the real facts, and this almost invariably leads to disaster.

5 To make this important point clearer, consider that an order for a fabric has been received, conditioned on a certain price. The records show that this fabric was previously made at a certain cost. It obviously would not do to accept the order without taking into consideration all changes in conditions that have taken place since the former production. Wages may have increased, automatic looms may have replaced plain looms, the price of raw material may have advanced, and a number of the elements of cost may have changed, which would alter the previous cost. Consequently an estimate is made, based on the known facts, taking these changes into consideration as accurately as possible. The order is accepted because the estimate indicates that, provided the conditions considered therein hold true, a profit should result from the sale. During the actual manufacture of the order, however, alterations of conditions occur which were not anticipated in the estimate. The production decreased, the shrinkage increased, the percentage of second quality goods increased; perhaps a number of such cost factors varied. In consequence the anticipated profit becomes doubtful and it is, therefore, evident that the estimate cannot be used as guarantee of profit. After the fabric is com-

[1] James Newton Gunn, "Engineering Magazine," January 1901.

pleted, the cost is carefully compiled and the true cost, regardless of the existing conditions, determined. This affords a proof of the accuracy or inaccuracy of the preliminary estimate, determines whether or not the order has been profitable and becomes a basis for estimating more accurately on future orders.

6 The second essential is predicated upon the first, that if cost is to be a statement of fact, it must be subject to proof. In order to insure this, it is essential that the person responsible for its compilation have full access to the most private information of the company's affairs and the unqualified confidence of its officers in order that every item of expense be included. Further, the accounting methods should be so arranged that the results obtained are carried to, and incorporated in, the commercial (private) books of the company.

7 The third essential is that the several departments involved work in complete harmony. All persons having to do with the recording and tabulating of statistics must work together for the single purpose of reporting information not alone accurately, but also promptly, in order that the data so recorded may be available before they become history. The sources of much of the information essential to cost must, of necessity, be from persons in the mill who are of a grade of labor lower than that which is responsible for the compilation of the costs, and, it is especially necessary, therefore, that they be impressed with the importance of the information they are recording.

8 The control of all materials not in process should be under the direction of a general storekeeper, and the control of production under a production manager.

9 It should be the function of the production manager to control production from the raw material to the finished product. Changes in production methods should be made only through the production manager and it should be his duty to keep the cost department fully informed at all times as to the methods in force and the quantities produced.

10 The general storekeeper should have charge of and be responsible for all raw material, semi-finished and finished stock wherever located. This does not necessarily imply that all stores should be centralized in one place, but it does imply that one person should be held responsible for their proper care and for correct reporting of the receipts and disbursements of all materials.

11 Manufacturers sometimes argue that because their material is

not liable to depletion by theft, from the lack of its value to an individual, they may safely figure their costs on the basis of a theoretical schedule of the material that should enter into any given unit of their product. Similarly, all indirect materials purchased are sometimes charged directly to expense and all direct materials to product, but this procedure is apt to prove embarrassing at times, as there will be an inventory of material on hand which theoretically has no value because it has already been charged to the product. Moreover, this practice is not conducive to accuracy as purchases of material are not necessarily made according to the manufacturing need for this material. It is usually found that savings may be made by buying in quantities sufficient for several months' normal operation of the mill.

12 Total cost may be divided into three main parts or divisions: (*a*) manufacturing or mill cost, (*b*) selling cost, (*c*) administrative cost. The latter two are generally known as commercial expense and are distinct from manufacturing or mill cost. They are usually combined and added to the manufacturing cost on a single percentage basis. While this is reasonably correct in a great many cases, it should be avoided wherever it is possible to obtain a more equitable basis. Many manufacturers dispose of their product through a selling agency or commission house, and therefore know exactly the selling expense chargeable to each fabric, and should accordingly apply this selling expense directly to the product.

13 Manufacturing cost is the sum of the following: (*a*) direct material cost, (*b*) direct labor cost, (*c*) indirect cost or burden.

14 Direct material cost includes the cost of all material entering into the product and directly assignable to it, and direct labor cost the wages paid which are directly chargeable to the product.

15 Indirect cost or burden includes the cost of labor and material and miscellaneous expenses not directly chargeable to the product, such as superintendence, power, heat, light, depreciation, etc. Speaking generally it includes all items not included in direct labor or materials.

16 Each of the above may be subdivided, if desired. The indirect cost, particularly, is a complex item and can be treated more readily if segregated for instance, into (1) power, heat and light, (2) departmental expenses, (3) general expense.

17 The cost of power, heat and light includes all expenditures incident to the production and distribution of power, heat or light. It is desirable to distribute the cost of power, heat and light by actual

consumption. Obviously more power is required to operate a heavy 72-in. loom than a light 28-in. one. A department using steam for manufacturing and heating should be charged proportionately more for steam than a department using it only for heating. The proper percentage may be obtained by actual measurement or it may be approximated from tests. While the former method is preferable, the latter is sufficiently exact if proper facilities are had for making accurate tests.

18 Departmental expense includes all items of expense which can be charged directly to a department, for instance, departmental supplies, foreman's wages, fixers' wages, etc. It can be apportioned to the product by one of a number of methods.

19 General expense includes all items of expense which are not directly chargeable to a department, and which therefore must be apportioned to the departments on a more or less arbitrary basis.

20 There are two general methods of collecting cost data: (*a*) by definite lots or orders; and (*b*) by operations. As the name implies, the first method provides for the determination of the cost of definite quantities of product and is applicable only to cases where the material can be processed in definite lots and kept intact through the several operations. These requirements render this method impracticable in such textile mills as largely manufacture what are known as staple products. The production in such cases is so continuous that it cannot be readily segregated into lots or batches. The plan is therefore applicable only to what are known as specialty or fancy mills in which it is the general practice to process the material in lots of comparatively limited quantities.

21 In order to insure that all expenditures incurred are included in the cost, it is necessary that all labor performed and material consumed should be charged to some form of order: Production orders for all labor and material expended on product intended for sale, plant orders for all expenditures for plant improvement and extension and standing orders for all expenditures that cannot be charged directly to either production or plant orders.

22 Production orders should be issued as the authority for processing all materials entering directly into the product. They should be numbered serially and all expenditures of labor and material on account of same should be charged to these numbers. It should be noted that it is not essential that an order should cover the complete process from raw material to finished product, but it is necessary that it should cover a complete stage of production.

23 Plant orders, like production orders, should be numbered serially and should serve the purpose of collecting the various items of labor and material expended on plant improvements and extensions.

24 Standing orders should be issued for collecting all expenditures of labor and material necessary for the efficient operation of the mill except those chargeable to production and plant orders and they will, therefore, embrace all indirect cost items. By means of standing orders, a minute and careful periodical comparison may be had of all manufacturing, administrative and selling expenses.

25 All disbursements of materials from stores should be priced at cost and charged to the proper order number. This will not only insure that all expenditures of material are accounted for, but it will also provide a means of collecting individually the material cost of the several orders.

26 There are several methods in use for collecting labor cost data. It has been in the past quite generally the practice to allow the workman to make his own record, or have his foreman or overseer do it for him. A better plan is to place the responsibility upon timekeepers, whose sole duty it is to collect labor information. Timekeepers should be supplemented, so far as possible, with mechanical recorders which can neither err nor misrepresent.

27 After obtaining the direct material and labor cost, the next step in the tabulation cost is to apply the indirect cost or burden. We shall not attempt to discuss at length the various methods of distributing indirect cost. The problem is much simplified if the departments are so defined that only similar operations and processes are contained within a single department. Distribution of expenses may be divided into two parts: first the distribution of the general expense of the mill as referred to above, together with power, heat and light to the producing departments; and second, the apportionment of the total departmental expense thus obtained over the product passing through the departments.

28 The first is not, in reality, extremely difficult, but it does require a certain amount of patience and study to determine the most equitable method of distributing the several items making up this general expense. Power, for example, as mentioned previously, should be distributed on the basis of power consumed, depreciation on the basis of the value of the plant and equipment used by the department. In a similar manner each item of general expense should be taken up, and the most equitable method of distribution determined.

29 The total departmental indirect cost thus obtained may be apportioned over the product by any one of several methods. It is quite generally desirable in textile mills, however, to use the machine hour basis or its equivalent, as the machine is largely the unit of production.

30 It will be readily seen that the cost of direct material and direct labor can be determined as soon as the order is completed, but that the exact amount to be added for indirect expense cannot be ascertained until after the close of the period.

31 It is, therefore, common practice in figuring current costs, to use the rates based on the results of previous periods. While this is the only practicable method to pursue when immediate cost figures are required, it, nevertheless, is liable to error and should only be used when it is not practicable to wait until the close of the period.

32 The second method, that of operation costs, provides, as the name implies, for the determination of the cost of individual operations. It is self-evident that, given the total cost of each operation and the loss by shrinkage of material from operation to operation, the total cost of any product, or the cost at any desired stage of completion may be found by combining these costs in the proper manner. They must, however, be combined only with a full knowledge of the shrinkage and the exact order in which the several operations were performed.

33 This method is applicable to mills in which production is continuous, and, therefore, is adapted to a large class of textile mills, especially those manufacturing staple cotton goods.

34 The foundation of operation costs rests upon an accurate knowledge of the production of each operation, or, stated another way, of the shrinkage from operation to operation. There are in use several methods of measuring production, but none of them are entirely satisfactory.

35 Because of the well-known fact that the majority of the fibers used in the manufacture of textiles are subject to a considerable variation in weight, due to the rapidity with which they absorb and discharge moisture, if weight is used as the unit of measure it is subject to a corresponding variation. While lineal measure would solve this difficulty it would so complicate the recording of the quantities of stock in process as to make the plan impracticable. Everything considered, the most practicable unit of measure for yarn, both finished and semi-finished, seems to be the pound. Cloth may be measured either by the pound or the yard, whichever is the more convenient.

36 As it is impracticable actually to weigh the product after each operation it is necessary to resort to some mechanical method of measurement. The most common method is the use of measuring clocks. Clocks are subject to possible error in that they measure the production based upon a fixed standard of efficiency, while it is, of course self-evident that any individual machine will fluctuate in its efficiency due to a number of causes.

37 The collection of the cost data under this plan differs from that used in the first plan, in that there is no special production order number against which the various expenditures of labor and material can be charged and it is, therefore, necessary to use the operation as the unit of cost. It is essential to this scheme that the various operations be definitely determined and designated by a number. The several different kinds of product should also be numbered.

38 A special ledger for collecting the cost data should be furnished in which provision should be made for collecting the direct material costs separately from the direct labor and the indirect cost.

39 All disbursements of direct material from stores should be priced at cost and charged in the mill ledger to the first operation through which the material passes, classified to kind of raw material. Provision should be made for crediting the several raw material accounts with the value of the waste made in each operation. The direct material accounts should be subdivided as to kind of raw material, rather than kind of product, as it is usually impracticable to obtain a report of waste classified as to kind of product. At best, the obtaining of accurate reports of waste made is a difficult proposition, especially in a mill where several kinds or grades of raw material are being processed along the same general lines, and quite possibly in the same rooms. It is important that this information be obtained as accurately as possible, in order that each operation and each kind of material may be credited with the proper amount for waste. The customary practice has been to consider that the shrinkage is the same for all kinds and grades of raw material and product. Shrinkages, on the contrary, are constant only in so far as the cause is either natural or due to mechanical appliances, and shrinkages due to "human elements" are subject to wide variation. Considering that the waste in woolen mills is over 50 per cent of the raw material and in cotton mills over 15 per cent, it seems well worth while to give this problem serious consideration. A mill agent once pointed to his waste wagon and aptly remarked, "there go our dividends." Apparently in many cases dividends might be

increased if more attention were given to the reduction and utilization of waste.

40 All direct labor should be analyzed as to operation and kind of product and charged in the mill ledger to the proper account.

41 The total indirect cost for each operation should be determined in much the same manner as explained under lot costs. It should then be apportioned to the several kinds of product and charged to the proper accounts in the mill ledger.

42 At the close of any period, the unit cost of any operation may be determined by dividing the total expenditure on the operation for the period, as shown by the mill ledger, by the total amount of product which has passed through the operation during the same period.

43 The total cost of any given product cannot be determined simply by adding together the several operation costs, considering the shrinkage from operation to operation. To obtain the cost of any operation in the completed product, it is necessary to increase the operation cost, in reciprocal proportion to the shrinkage from the end of that operation to the finished product. For example, if the cost of an operation is \$0.015 and the shrinkage of product is 25 per cent, the operation cost in completed product would be $\frac{\$0.015}{100\% - 25\%} = \$0.02$. It would be possible to determine the total cost of product by this method, except for the fact that the exact sequence of operations is not readily ascertainable, making it impossible to determine accurately the amount of shrinkage, especially in a mill making a variety of products.

44 A more satisfactory method is to build up the total cost, operation by operation, beginning with the first operation and charging forward to each succeeding operation the cumulative cost of all preceding operations, inventories, of course, considered. This plan has the advantage of following the product from operation to operation, according to the manufacturing layout and schedule.

45 It is obvious that the output of an operation plus the waste made is equal to its input. Providing no inventory exists between this and the succeeding operation, the good output of the first operation is equal to the output of the second, including the waste made. Ordinarily, however, inventories do exist and must, therefore, be taken into consideration. An equation taking inventory into consideration may be made up as follows: The good output of the first operation, plus the inventory of the product of that operation at the beginning of the

period, less the inventory of that operation at the end of the period, is equal to the input of the succeeding operation. If it were practicable to determine the input of each operation, it would be possible by means of this formula to determine the inventory on hand between operations, at the end of any period; but in most instances there seems to be no practical method of obtaining the input of each operation. Consequently, a physical inventory of the product in process must be taken whenever the inventory is desired. With the inventory a known factor, by means of the above equation the quantity and value of the product to be charged forward from operation to operation can be easily determined. Each successive operation should be charged with the total cost of all preceding operations, inventories considered, and by this means it thus becomes possible to determine the accumulated cost of any product at any stage of completion.

46 The accumulated unit cost may be obtained by dividing the accumulated cost at that stage of production by the output of the operation. This unit cost should be used in pricing the physical inventory of the product from that operation.

47 In many cases, the product of one operation is made into a number of different kinds of product in a succeeding operation. For example, one size of roving may be spun into several sizes of yarn and it is necessary in such cases to apportion the amount charged forward from the previous operation to the succeeding operation, based on the total production of each kind of product in the succeeding operation. It will be evident that by this method of cost finding, the inventory of product in process is automatically priced and, further, that it is possible to determine the cost of the product shipped during the period. The operation cost method not only provides for obtaining the detail costs by operations, but also the total cost of each kind of product.

48 Cost accounting is most valuable from the standpoint of the manufacturer, in that it shows numerous ways for reducing operating costs. It is self-evident that accurate costs are of fundamental importance as a guide in determining whether to meet or withdraw from unintelligent competition. They are also essential in the preparation of financial statements, as a knowledge of costs is absolutely necessary to the proper valuation of assets, and for the making up of a profit and loss account.

49 As heretofore mentioned, a mill or inventory ledger should be opened, designed to collect and control the various facts regarding

the manufacturing costs and inventory records. This should be done regardless of whether costs are determined by means of the definite lot or operation plan. The mill ledger should be controlled by the private ledger and at the close of each period should be in balance with the controlling account in the private ledger.

50 The various items of expense not distributed or applying to product in process, should be so segregated that they can be easily checked against uncompleted costs and the uncompleted costs in turn checked against the product in process account. By this means at the close of each period, be it one or four weeks, or one of the calendar months or several months, the accuracy of the costs compiled during the period is demonstrated and when so demonstrated is practically beyond question, as the accuracy then reverts to the weight or count of raw material.

No. 1409

# EFFICIENCY OF ROPE DRIVING AS A MEANS OF POWER TRANSMISSION

By E. H. Ahara, Mishawaka, Ind.

Member of the Society

Of the several types of mechanical power transmission including belting, gearing, rope driving, etc., the question of efficiency of the last has perhaps received the least attention and, outside of general treatment, very little engineering literature, considering the importance of the subject, is obtainable. While transmission by rope systems of various kinds is a very old method of transmitting power from one shaft to another, having been used, according to one authority, to a limited extent by C. F. Hern, of Logleback on the Rhine, Germany, as early as 1852, the difficulty of obtaining accurate measurements of the power consumed by the rope itself has prevented its efficiency being easily ascertained in practical work. Because of the ease with which electrical measurements can be taken by means of direct reading instruments, which enabled power consumption to be measured so much more readily than by any other means, electrical power itself has been given much more attention than other types of power service.

2 Owing to there being no way of determining the losses in transmission in regular working rope drive installations, the efficiency can be ascertained only by laboratory, or test outfits, and these, considering the diversity of types and methods, the large number of variables involved, as well as the vast difference in working conditions and capacities, are necessarily elaborate and the expense of such tests has tended to limit the knowledge obtainable in this way. The present tests were undertaken with the object of obtaining, in a practical way, some definite figures on a few of the most important of the variables in rope-driving practice, and it is believed that the plan is broad enough in its scope to give reliable data whereby the efficiency of most drives may be very closely approximated.

3 The importance of the tests may be estimated from the conclusions that are deducible from the data obtained. They seem to show that the efficiency in rope driving is considerably greater at the lower

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

speeds than at the higher ones, the dropping off being especially noticeable above 4500 ft. per min. of rope speed. They also show that the efficiency of a rope drive is not materially affected by distances between centers up to 150 ft., which was the limit of the tests, that the drop of efficiency at 50 per cent load is comparatively small over that of full load, and that if proper care is exercised to have all grooves perfect in pitch diameter many as well as few ropes can be run on a drive with good efficiency. With the American system, increasing the slack rope tension up to 360 lb. in a 1-in. rope does not appear to decrease its efficiency, but rather to increase it if power is used in conformity with this tension; but only such tension should be used as is necessary to drive the load needed. For the rope tested the American system has very much more capacity than the English system, and has also a higher percentage of efficiency. In general it would appear that, where there is considerable power to be transmitted, the properly worked out rope drive gives a most efficient and economical method, and where conditions are favorable to its installation no other known method of transmission will so well conserve power losses.

4 There are two general systems of rope driving in common use: (*a*) the English system, the oldest, which uses a series of separate ropes, each spliced up into an endless band and occupying a separate groove on the face of each of the sheave wheels; and (*b*) the American system, the more modern, patented in the United States by W. H. Dodge, June 23, 1885, which uses one continuous rope wound about the driver and driven sheaves with a loop taken over a weighed tightener wheel, that keeps a definite tension continually in the ropes. Wire ropes or cables are sometimes used, but so rarely that no attempt was made to study their efficiency. Manila fiber ropes, which are used almost entirely in rope transmission work, were used exclusively in these tests.

5 These two general systems were tested in open drives of from one to eight ropes each under speeds of rope travel from 2500 ft. per min. to 5500 ft. per min., on center distances varying from 25 ft. to 150 ft. and with varying loads. Difficulty was found in handling the English system on 125 ft. and 150 ft. centers on account of the slack rope, which was the upper one, sagging down through the tight ropes and rubbing on the ground. This limited the work on the English system to 112-ft. centers as the maximum. Other than this limit, and the fact of varying tensions on the 100-ft. centers, the English test followed the same schedule as the American.

6 Both English and American systems were also tested on what is termed "up and over" drives, i. e., with four idlers, on approximately 100-ft. centers with the same approximate speeds as above, and with from one to eight ropes. There were also tests taken on the American system to determine the efficiency and capacity under different strains in the pulling rope. In all about 700 tests were taken with upwards of 7000 readings. The tests were all taken in the open air, at the plant of the Dodge Manufacturing Company, Mishawaka, Indiana, and extended over a period of five months of continuous work, ten hours a day, from August to December 1912. But few days were lost during the entire series of tests and these only on account of rainy weather.

7 The general method of conducting the test embraced the use of a 250-h.p. Westinghouse direct-current motor driving through an auxiliary rope drive to a jack shaft on which was mounted the 60-in. test driver; the driven shaft was fitted in bearings on a movable tower, and on the driven shaft was mounted the receiving sheave, of the same diameter as the driving sheave, and also a prony-brake wheel. Fig. 1 is a view of the equipment in operation with sheaves at 50 ft. centers and carrying seven ropes at 4500 ft. per min. on the American open-drive system, at a time when 155 h.p. was being transmitted. The test outfit included also a Weston standard volt meter, a Westinghouse milli-volt meter with 750 ampere shunt, a Schaeffer and Budenberg hand tachometer with three scales, two hand revolution counters, and a standard Fairbanks platform scale.

8 In taking a test one observer recorded the electrical consumption, the polarity being reversed and the average voltage taken; another applied the brake and kept the load constant; a third recorded the revolutions of the driven shaft; a fourth recorded the revolutions of the driver, and still another observed the sag of the rope in the center of the span. By means of electrical signals all observations were started at the same instant to avoid the effect of continual slight variations; if there arose any reason for doubt of a test being correct it was repeated. The ammeter used was compared weekly with a carefully calibrated test instrument and found to maintain its accuracy during the test. The voltmeter was frequently checked by another constantly in circuit on the main switchboard. The start was made by means of a water rheostat which was cut out by a knife switch when the motor reached full speed.

9 Owing to the large variations needed in loading on the prony

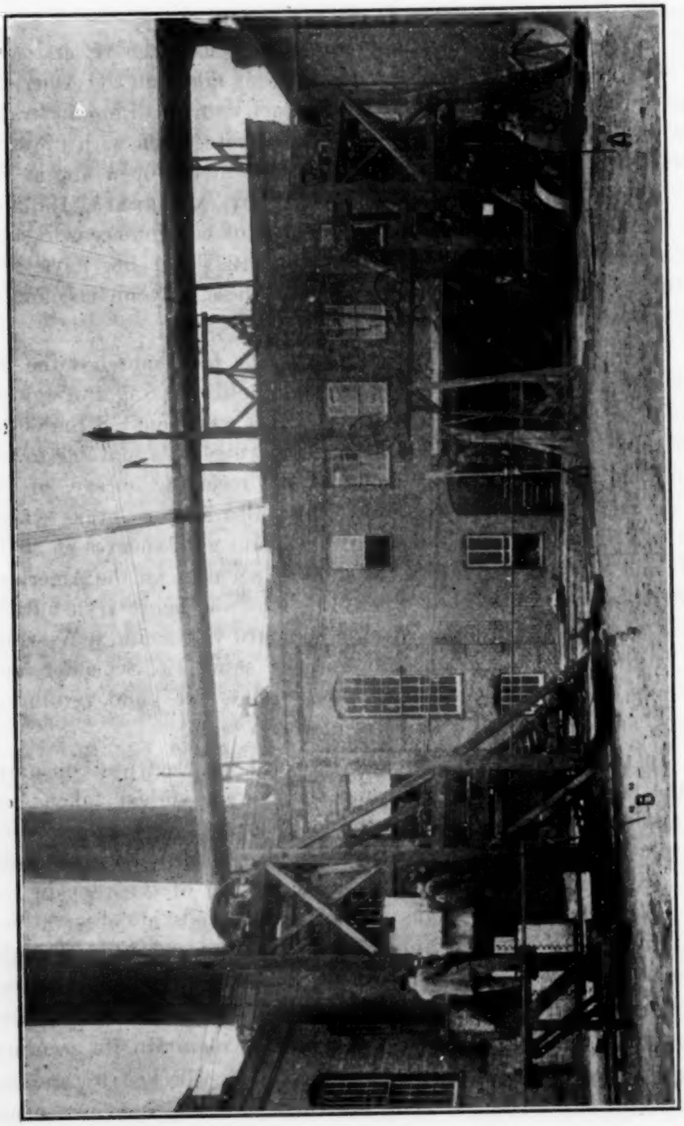



Fig. 1 Arrangement of Apparatus, American Open Drive, 50 Ft. on Centers, Seven Ropes, 4500 Ft. per Min., 155 H.P.

brake the load measurements were taken directly on a platform scale, careful corections being made for the weight of the prony-brake lever; also the latter was made of such length that $\frac{\text{Wt.} \times \text{r. p. m.}}{1000} = \text{h. p.}$ The brake wheel was flanged so as to hold a considerable

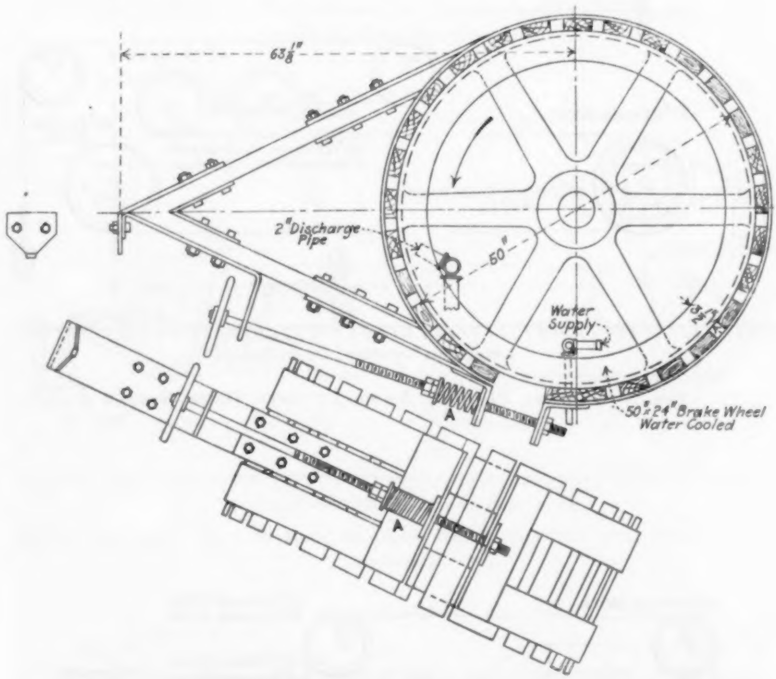



FIG. 2 DETAILS OF PRONY BRAKE USED IN TESTS

quantity of water on the inside of the rim while running, and cooling water was constantly supplied from a city water main, through a 1-in. pipe, and carried away through a 2-in. pipe; the quantity used being varied by means of a valve to suit conditions. The surface of the brake wheel was well lubricated, and having a brake band faced with cross bars of hard maple, no particular difficulty was experienced in working the brake up to 250 h.p. A spring used in the brake pressure connection at *A*, Fig. 2, prevented sudden seizing of the brake band and aided in keeping the scale load uniform. By using a proper

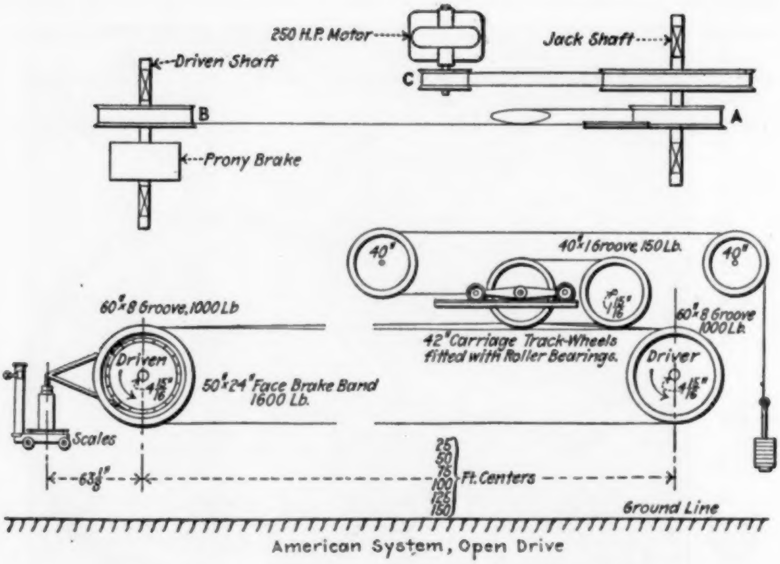



FIG. 3 GENERAL PLAN OF AMERICAN OPEN DRIVE SYSTEM

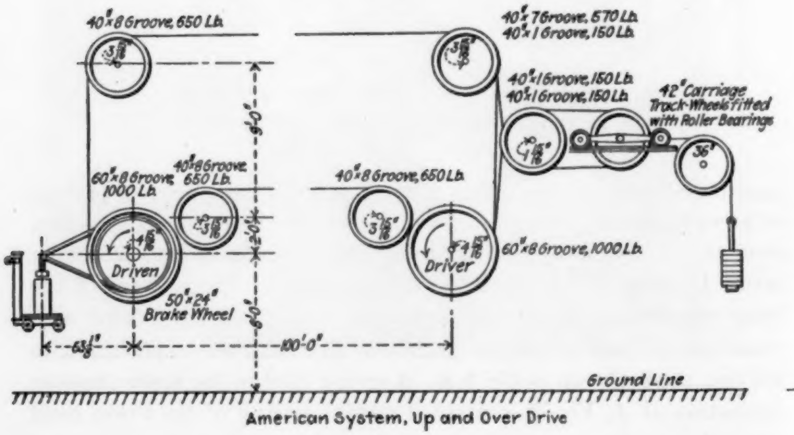



FIG. 4 GENERAL PLAN OF AMERICAN "UP AND OVER" DRIVE SYSTEM

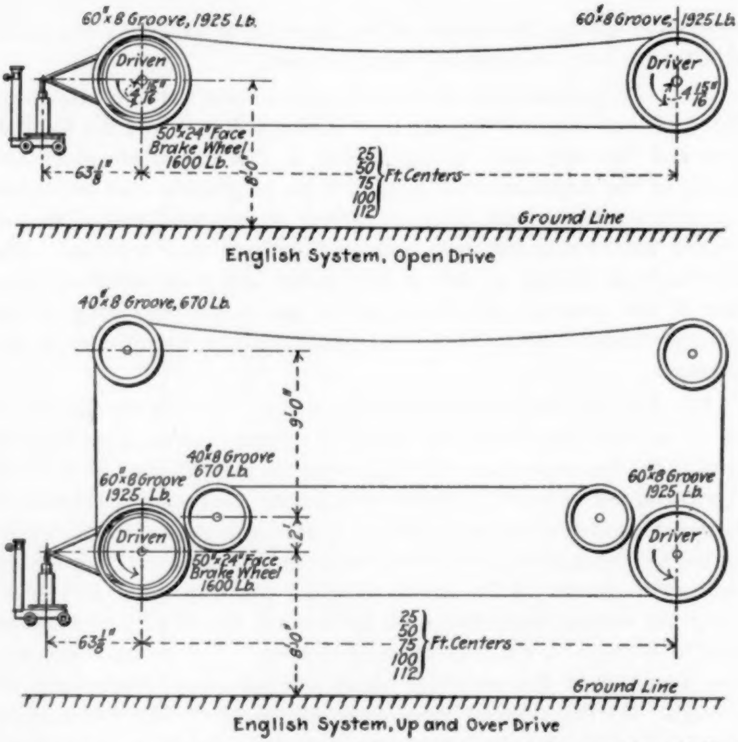



Fig. 5 General Plan of English Open Drive and "Up and Over" Drive Systems

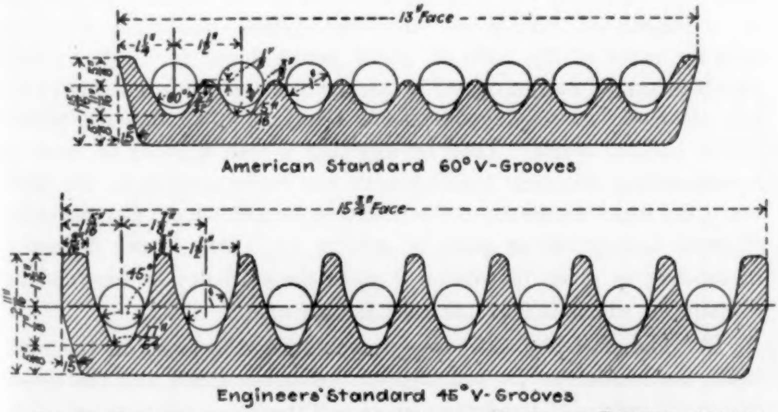



Fig. 6 Details of American and English Types of Grooves used on Driver and Driven Wheels

capacity of spring, the tension of which could be varied by the screw adjustment, very good results were obtained.

10 The general plan of the American open drive is shown in Fig. 3, of the American "up and over" drive in Fig. 4, and the English open and "up and over" drives in Fig. 5. In Fig. 6 are given the details of the American and English types of grooves used on driver and driven wheels. All idler or carrier wheels used had U-shaped grooves into which the rope bottomed without side friction. The weights of all wheels, as well as dimensions, are given to afford some idea of the inertia. All bearings on all shafts were ring oiling bearings except those on the carriage axle which were ordinary grease bearings.

11 The different speeds were obtained by changing the sheaves on the motor and jack shafts and for convenience sheaves were used in pairs that gave respectively 2500 ft., 3500 ft., 4500 ft., and 5500 ft. of rope travel per minute. In order to eliminate the factor of friction of the motor itself and the jack shaft, with intervening drive, the tests were started by first applying the prony brake to the jack shaft, and taking readings to determine the friction load of the motor and jack shaft under the various loads and with the various speeds of jack shaft as were to be used in the rope-drive tests later. The prony brake was then removed to the receiving shaft and the tests proper carried through. At the close of these observations, the brake was applied to the jack shaft and the friction readings again taken for comparison with the friction readings taken at the beginning of the test. These data were tabulated and charted, a sample of the tabulations for one set of conditions, namely the American system, open-drive, operated with six ropes at the 2500 ft. speed, being shown in Table 1; this gave a measured output of 945 lb. on the brake at 156 r.p.m., or 147.4 h.p., the electrical readings being 262 volts and 508.5 amperes, which equals 133,227 watts. Then by applying a load such as to cause a corresponding electrical reading with the brake applied to the jack shaft, the input to the drive was found to be 153.4 h.p. Comparisons of input horsepower as given in column 13, Table 1, and delivered horsepower as given in column 9 gives the efficiency as recorded in column 14, which is 96.1 per cent in this case.

12 The efficiency as above given includes the losses in the rope itself, the friction in the bearings on the driven shaft, and the losses due to the inertia of the driven wheel and the prony-brake wheel, and, also, in the American system the friction of the tension equipment.

TABLE 1 METHOD OF TABULATION OF DATA

TEST ON ROPE DRIVE EFFICIENCY

AMERICAN SYSTEM—OPEN DRIVE DATE, 9-30-12

| No. of Ropes | Speed of Ropes | Wt. on Take-up | Sheave Centers | Sag of Rope | | Prony Brake | | | Electrical Reading | | | Net H. P. Consumption Compared with Prony Brake | Efficiency | Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Tight | Loose | Weight | Revolutions | H. P. | Volts | Amperes | Watts | | | |
| Col. 1 | Col. 2 | Col. 3 | Col. 4 | Col. 5 | Col. 6 | Col. 7 | Col. 8 | Col. 9 | Col. 10 | Col. 11 | Col. 12 | Col. 13 | Col.14 | Col. 15 |
| | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | |
| 6 | 2500 | 360 lb. | 50′-0″ | 3″ | 7″ | 945 | 156 | 147.9 | 262 | 508.5 | 133227 | 153.4 | 96.1 | |
| | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | |

All tests were made to include the friction of the receiving shaft and bearings as it was thought this would more nearly approximate working conditions and make the data obtained of more general application in ordinary comparisons. All bearings used were of the ring oiling babbitted type, and the ropes were all 1-in. manila rope of best quality, carefully treated with a rope dressing to prevent the entrance of moisture and to keep the surface in as nearly uniform

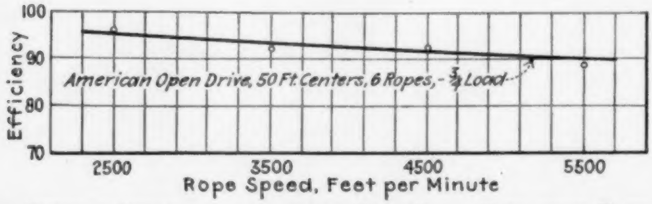



FIG. 7 RESULTS OBTAINED IN AMERICAN OPEN DRIVE TEST WITH EFFICIENCY PLOTTED AGAINST SPEED

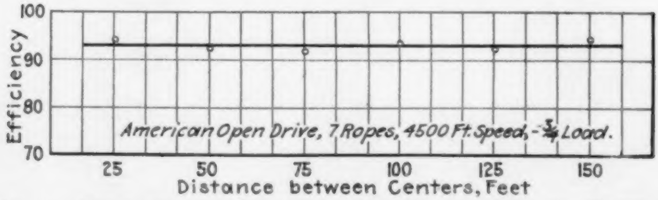



FIG. 8 CURVE SHOWING EFFICIENCY RELATIVE TO VARYING CENTERS

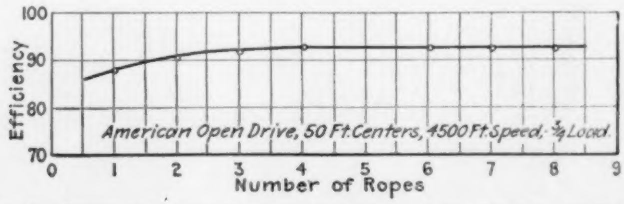



FIG. 9 CURVE SHOWING EFFICIENCY RELATIVE TO NUMBER OF ROPES

condition as possible. It was found during the test that this was of great importance as very slight changes in the rope surface immediately affected its capacity and efficiency.

13 The same rope was used throughout the American open drive test, the tests being run in such order that the rope was shortened at each succeeding test, and thus there was never more than one splice in the rope, thereby reducing any inaccuracy from lack of uniformity

in the splice of the rope to the least possible point. This same care was observed in all of the tests, no rope being used with more than the necessary single splice, and these were in every case carefully made. Frequent photographs were taken showing the positions of the ropes under varying conditions, speeds and centers, which gave a very good idea of the test as it proceeded.

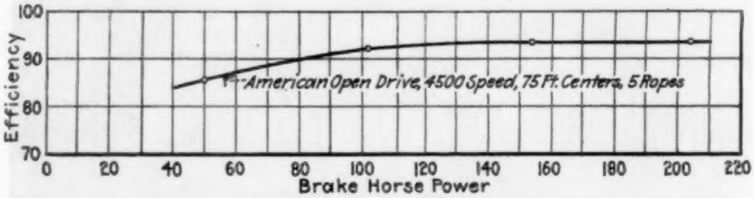



Fig. 10 Curve showing Comparison of Efficiency with Various Loads

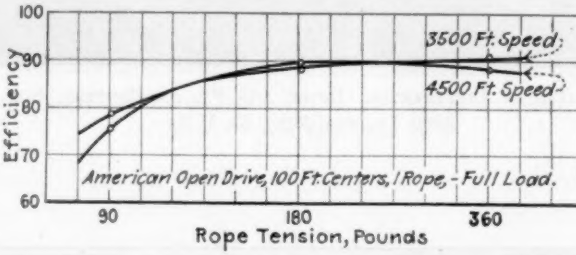



Fig. 11 Curve showing Efficiency plotted against Various Rope Tensions

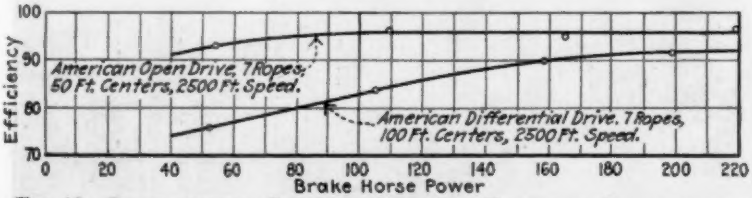



Fig. 12 Curve showing Comparison between Exact and Differential Drives

14 Fig. 7 gives the results obtained in the American open drive test for six ropes 50-ft. centers with efficiency plotted against speed; Fig. 8, the results of efficiency relative to varying centers, seven ropes at the 4500 ft. speed, and Fig. 9, efficiency relative to number of ropes for 50-ft. centers at the 4500 ft. speed.

15 It will be seen that a lower efficiency is shown on the lower number of ropes. This is undoubtedly caused by the fact that the

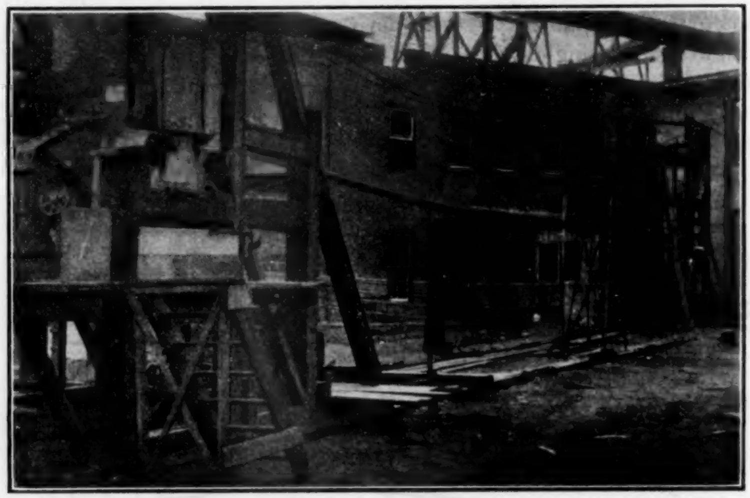


FIG. 13 AMERICAN DIFFERENTIAL DRIVE, 100 FT. ON CENTERS, SIX ROPES, 2500 FT. PER MIN., 55 H.P.

FIG. 14 AMERICAN DIFFERENTIAL DRIVE, 100 FT. ON CENTERS, SIX ROPES, 2500 FT. PER MIN., 110 H.P.

FIG. 15 AMERICAN DIFFERENTIAL DRIVE, 100 FT. ON CENTERS, SIX ROPES, 2500 FT. PER MIN., 165 H. P.

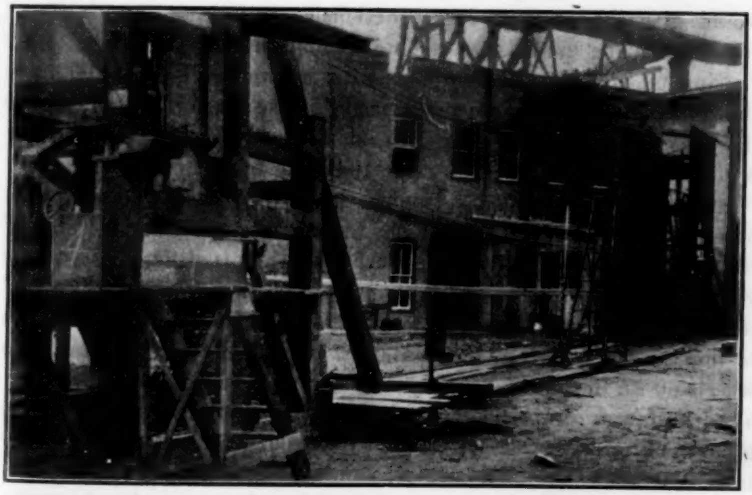


FIG. 16 AMERICAN DIFFERENTIAL DRIVE, 100 FT. ON CENTERS, SIX ROPES, 2500 FT. PER MIN., 220 H. P.

eight-grooved wheel made for testing eight ropes was used for all the lesser number of ropes to avoid the expense of making additional wheels with a suitable number of grooves for each drive. This together with the weight of the prony brake increased the proportional friction on the smaller loads, and is, therefore, a larger percentage of the total load in the drive of few ropes. With the friction of driven shaft eliminated, limiting the losses to those in the rope alone, it is believed the efficiency in a single rope would be as great if not greater than any multiple thereof.

16 All readings were taken at four different capacities. The rope was brought up to speed, and the brake gradually applied until the observation of r.p.m. on the driving and driven shafts showed a slippage, when the brake pressure was released a trifle and the observation called "full load" taken. By changing the brake, 75 per cent, 50 per cent and 25 per cent of this load was also observed. Fig. 10, which is a good representative case, gives the comparisons of efficiency with these various loads. Owing to the limited capacity of the motor (250 h.p.) the slippage point was not reached on many of the tests involving the higher speeds and number of ropes. In these cases the high record was taken at the motor capacity and so recorded.

17 In all of the above tests the tension in the slack side of the rope was kept as uniform as possible at 180 lb. by means of weighting the tension carriage properly. In order to ascertain what tension would give the greatest degree of efficiency, irrespective of life of rope or wearing qualities, a test was made with the slack rope under varying tensions, the results of which are given in Fig. 11. This is shown on one rope because the capacity of the single rope was, under the heavy tension, as great as that of the motor, so no proper comparison could be obtained on more than one rope.

18 One of the greatest troubles in rope driving, when installed by experienced engineers, has been the lack of uniformity in pitch of the grooves where many ropes were used. This differential in the grooves compels slippage of the ropes which not only causes loss of power, but also rapidly depreciates the rope, and often causes the rope to flop around badly. To test the loss of efficiency in this case two wheels 60 in. in diameter, of eight grooves each, were made up exactly alike so far as could be, and after most careful measurements the grooves on either wheel were found not to differ more than 1/64 in. in circumference from any other on the same sheave, and a test made. The driver was then removed, and the pitch diameter of each groove made approximately 1/32 in. less than the preceding groove so the

eighth groove was $\frac{1}{4}$ in. less in diameter than the first one. Again it was placed on the driving shaft and a duplicate set of tests made, the general tendencies of which are given in Fig. 12. In this test as the first groove was $\frac{1}{4}$ in. larger in pitch diameter than the eighth groove, with the sheave revolving at 160 r.p.m., there would necessarily be a slippage in an inelastic band of over ten feet per minute.

19 The photographs taken of this differential drive, shown in Figs. 13 to 16, in which the horsepower transmitted is 52 h.p., 110 h.p., 165 h.p., and 220 h.p., are quite illuminating, as they show plainly the sag of the various ropes under this condition, and how under heavy load they stretch and slip on the grooves of the sheave. The elasticity of the rope in this case undoubtedly lends itself to aid efficient operation.

20 The "up and over" American drive, so called because of its course upward from a driver, over idler wheels, then horizontally for a distance and down over idler wheels to the driven wheel, was tested because it is typical of all but direct drives, i.e., straight from driver to driven, and the difference in efficiency between this type and the open drive, as shown in Fig. 17, will serve as a basis of comparison where but two additional right-angle turns are introduced requiring four additional idlers. This test was run on approximately 100-ft. centers with a new rope, the same rope being afterwards used in the testing of the various English drives.

21 In testing the English drives, that proper comparisons might be made, the ropes were all cut to the same measured length, and the splices carefully made, so that all ropes might be equally tight when applied to the sheaves. The movable tower was then carefully moved back, the rope being run slowly meanwhile until the sag in the rope was approximately the same as in the American drive of the same center distance in which the slack rope was under a tension of 180 lb. This is undoubtedly a higher tension than is used in general practice on the English drive system with this size of rope, and the capacities and efficiencies shown are perhaps higher than are ordinarily attained in common practice. Fig. 18 gives the efficiencies, when compared by speeds, of the English open drive. This shows a marked decrease in efficiency as the speed of the rope increases, falling as low as 84.5 per cent at 5500 ft. rope speed.

22 The general tendency in all tests made, both English and American, was corroborative of this tendency toward decrease in efficiency as the speed of the rope increases. There was, however, a lack of smoothness to many of the curves owing, it was thought,

to variations in rope tension, and perhaps slight changes in the surface of the rope due to climatic changes. This was very much more noticeable on the English than on the American systems, the irregularity of the curve on the former being affected by the fact that a greater load was sometimes attained on a certain speed than could later be reached on the next higher speed. The results relative to varying centers with six ropes on the 4500 ft. speed are given in Fig. 19, and

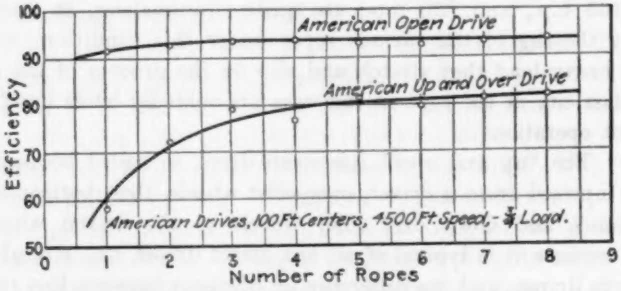



FIG. 17 CURVES SHOWING DIFFERENCE IN EFFICIENCY BETWEEN AMERICAN OPEN DRIVE AND AMERICAN "UP AND OVER" DRIVE WITH VARYING NUMBERS OF ROPES

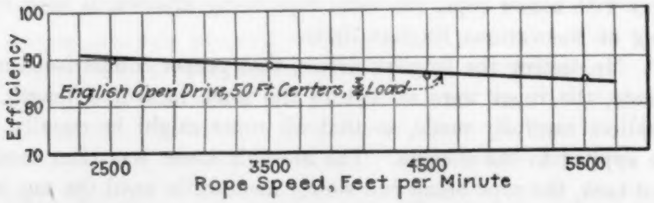



FIG. 18 CURVE SHOWING EFFICIENCIES AT DIFFERENT SPEEDS WITH ENGLISH OPEN DRIVE

Fig. 20 gives the results relative to number of ropes with 75 ft. centers and the 3500 ft. speed.

23 It will be seen by reference to Fig. 19 that the efficiency of the English drive, within the limit of distances tested, remains practically constant irrespective of center distance; while with reference to number of ropes, it increases as does the American for the first few ropes, and then remains practically constant through the additional ones. Fig. 21 gives the "up and over" English drive relative to number of ropes plotted on the four different speeds. This shows very clearly the greater efficiencies of the slower speeds.

The center distance on all the "up and over" drive tests was approximately 100 ft., so that direct comparisons with the open drives on the corresponding centers might be made.

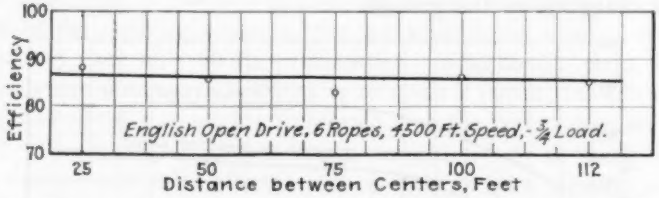



FIG. 19 CURVE SHOWING EFFICIENCIES OF ENGLISH OPEN DRIVE WITH VARYING CENTERS

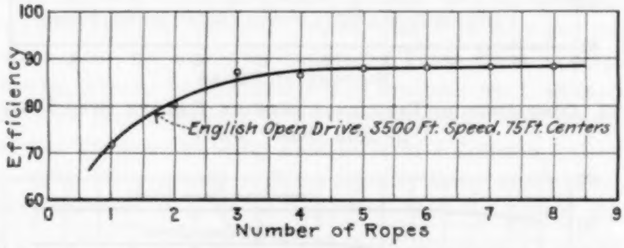



FIG. 20 CURVE SHOWING EFFICIENCY OF ENGLISH OPEN DRIVE RELATIVE TO NUMBER OF ROPES

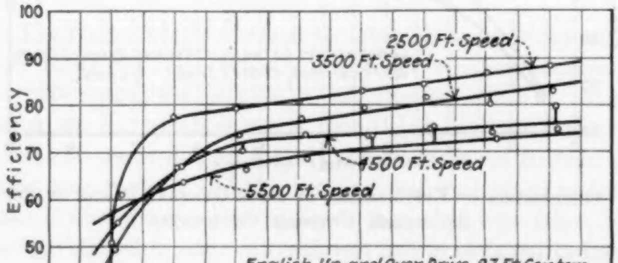



FIG. 21 CURVE SHOWING EFFICIENCIES OF "UP AND OVER" ENGLISH DRIVE WITH DIFFERENT NUMBERS OF ROPES AT FOUR DIFFERENT SPEEDS

24 In order to test the effect of tension on the English system a comparison was made by moving the receiving tower forward 1¼ ft. after making a series of tests, and then making another corresponding

series. When sheaves with proper English pinch grooves, as shown in Fig. 6, were used there was no appreciable difference either in power transmitted or in efficiency. Greater variations than $1\frac{1}{4}$ ft. in center distances could not be satisfactorily tried because of the slack ropes dragging on the ground.

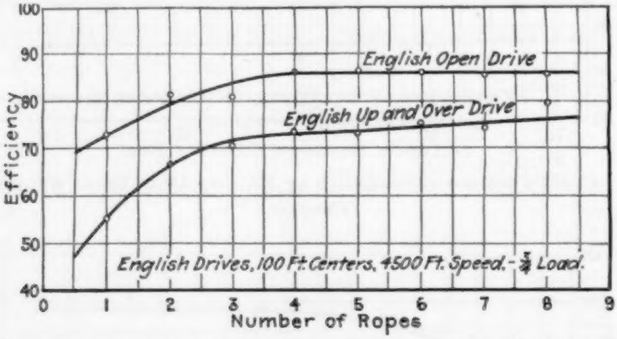



Fig. 22 Comparison of Efficiency between English Open Drive and "Up and Over" Drive

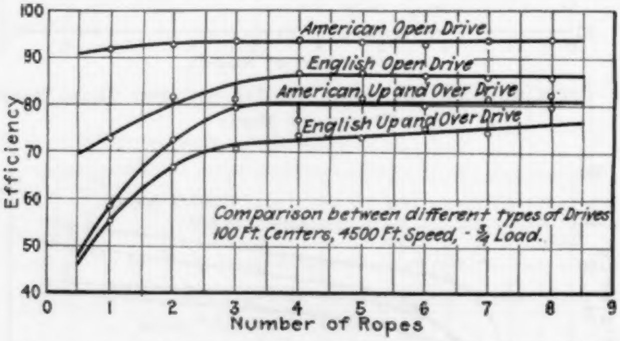



Fig. 23 Comparison of Efficiencies of Four General Plans of Rope Driving under Uniform Conditions

25 A comparison of efficiency between the English open drive and the English "up and over" drive is given in Fig. 22. In Fig. 23 is given a representative comparison of the efficiencies of the four general plans of rope driving on a center distance of 100 ft. and a rope speed of 4500 ft. per min.

26 A new rope was used on the American open drive, another new one being started with the American "up and over," which latter was also used on both English drives. The time of introducing

the new rope is mentioned because it was found that a new rope had very much less capacity, owing to its low coefficient of friction, than the same one had after it had run a short time in work, and having less capacity it would usually have less efficiency also. The stiffness of the rope when it was new also seemed to add friction to the drive owing to the bending action in passing around the sheaves, which on the test were only 60 in. in diameter. With larger main drive wheels, where the rope is not bent to so small a radius, the efficiency would undoubtedly be slightly increased over the results here shown.

27 The capacities of rope drives are affected by so many variables that no particular attempt is here made to state capacity other than in general figures, obtained as indicated in the early part of this paper, and used merely to have a uniform method of taking the tests. Fig. 24, however, gives a general idea of some limiting capacities obtained in the open drives, and Fig. 25 some of those obtained in the "up and over" drives. These limiting capacities shown are in no sense to be considered as available working capacities, but merely as limits reached in driving capacities, the general tendency of the whole series being used to form a decision, rather than any one curve, the practical working capacity being much lower than the limits shown. It is to be regretted that the power available was not great enough to get the larger capacities as so few points were obtainable on the American open system. Even at the low speed of 2500 ft. four ropes could not be made to slip with the 250 h.p. available.

28 The irregularities of these limiting horsepower capacities are caused by many variables, some of which are weather, surface conditions of rope, condition of rope wheel grooves, etc. In connection with the groove in the wheel it was found that if a rope began to slip in a groove and warmed it up slightly the coefficient of friction seemed to decrease very rapidly, and if a limiting test were made under these conditions it was invariably much lower than if it were taken just before the rope started to slip. As each test had to be taken in its scheduled order some of those shown were taken at intervals of several weeks.

29 As the limiting capacities of the English drive are very much lower than the American, it is thought the efficiency is likewise affected, the friction in the former case being a much larger proportion of the power delivered. And again, the English system in order to get driving power in the rope, pinches it in the narrow groove requiring considerable force to withdraw it again. This effort being

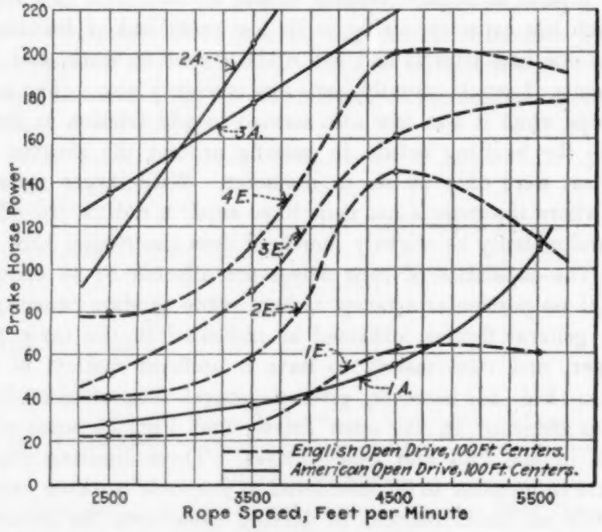



FIG. 24 CURVES SHOWING LIMITS OF CAPACITY OF TWO FORMS OF OPEN DRIVE AT FOUR DIFFERENT SPEEDS

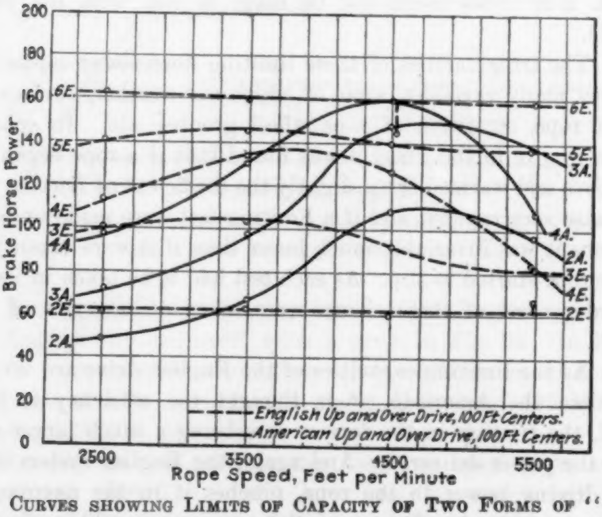



FIG. 25 CURVES SHOWING LIMITS OF CAPACITY OF TWO FORMS OF "UP AND OVER" DRIVE AT FOUR DIFFERENT SPEEDS

exerted continuously uses up considerable power. The American system, on the contrary, uses a 60-deg. groove in which the rope lies freely and withdraws without effort.

30 In conclusion, much credit is due Mr. W. H. Tupper who gave important aid in the electrical part of the work, and to Mr. E. M. Carver who personally superintended the test work and carried out the major part of the computations.

## DISCUSSION

Henry G. Stott said that the question of efficiency was, of course, a very important one, but that the question of reliability was also involved. He had had a good deal of experience in driving high-speed machines with rope drive in Europe, where both systems were used. While the American system of continuous rope was a good deal easier to operate, yet in the case of an accident to the rope, the entire rope became tangled up and something else was broken; whereas with the English system of separate drives, there were a number of ropes, each one separate, and if one rope broke the work was not interrupted, so that the question of reliability should also be taken into consideration.

Geo. N. Van Derhoef (written). Why does the efficiency decrease as the speed increases, in both the American and English systems? In the American system the slack side tension at any speed is equal to one-half the pull of the carriage plus the centrifugal force due to the speed; while the slack side tension in the English system is due to the initial tension less the centrifugal tension, and less a decrease in tension due to the total elongation of the rope band due to its elasticity. Is this due to "skin friction" of the rope sliding through the air, or to bending and unbending of the rope (either or both of which would indicate a functional loss due to speed), or on the other hand, is it caused by a change in the coefficient of friction of the bearings, which would indicate that the loss was entirely outside of the rope and that a high-speed rope may be as efficient as a low-speed one? This is quite an important point because in the design of many large drives the speed or travel of the rope can be made almost anything desired.

The tests showing the results for different distances between centers would seem to eliminate the "skin friction" idea, because the difference in the total length of rope is sufficiently marked to make

any loss of this kind of any amount quite perceptible. In the English system test, the slight downward tendency to the average line is not sufficiently marked so that any conclusion can be drawn.

The tests showing that with the American system the efficiency at half load is practically the same as at full load indicates that a drive may be designed to take care of occasional peak loads without a sacrifice in efficiency when operating at the average normal load we usually get in practice; or, to put it another way, without loss in efficiency when meeting a peak load, which, although generally of momentary duration, may at times be prolonged over considerable periods. If this is functionally true of American system rope drives in general, it is a most important characteristic.

The tests covering the efficiency with different numbers of ropes with drives of both systems show two important things: (*a*) that it is of basic importance that the sheaves be accurately grooved; and (*b*) that with properly grooved sheaves, drives for large powers, where many wraps of rope must be used, will not apparently be accompanied with any loss in efficiency. This seems to be quite different from what we find with large belt drives where the crowning of the pulleys gives an inherent loss that cannot be entirely prevented. A corollary from the above is that the choice between a few ropes of large diameter and a good many ropes of small diameter is not dependent upon the question of efficiency but on other design factors.

The fact that increasing the slack side tension up to at least 360 lb. for a 1-in. rope increases rather than decreases the efficiency is important both to the designer and user. No doubt the life of a transmission rope depends very largely on the load factor. Still, it is often necessary to increase materially the load on any transmission due to changes in operation, emergency, etc., far beyond that contemplated in the design. That this can be done with the American system simply by increasing the tension weight and at no greater penalty than a more or less proportionate decrease in the life of the rope means great flexibility and certainty in getting out production.

That the efficiency of the so-called up-and-over drive is less than the direct is only what might be expected from the greater number of parts. That this seems to be rather less marked in the American than the English system is in line with the results shown by all the various tests.

That the coefficient of friction of a new rope is less than that of a conditioned one is well known. In the design of an English system

drive great care must be exercised to have the drive amply large to transmit the power desired when the rope is new, otherwise full operation cannot be secured until the ropes have been conditioned. In designing the American system it is not necessary to give any attention to the coefficient of friction of a new rope at all. The design should be based on the actual normal running condition. All that is necessary to do when a new rope is put on the drive is to use the proper amount of tension weight to give the desired traction and to reduce this as rapidly as the rope becomes conditioned and the coefficient of friction rises to the normal point.

To secure all the advantages that are possible with the American system drive, the driving and driven shafts should be of ample size so that extra tension weight can be used whenever desired, otherwise its great flexibility would not be available on account of a detail that should have attention at the time of designing.

W. H. Kenerson asked whether the result, outlined in one of the curves, showing that the efficiency dropped with increase in speed, was derived from the four individual observations, or whether the points were the average of a large number. If, as the author further showed in another curve, the distance between centers did not affect the efficiency, or in other words, the line drawn through those five points was a straight line horizontally, it would be seen that if those were individual results only, it could be just as fairly deduced from the first curve that the line was horizontal; in other words, the variations due to other causes, possibly, would be as much as 4 per cent, as indicated.

Selby Haar (written). Having had occasion to estimate the losses of power in rope transmission, I can cordially endorse Mr. Ahara's opening statement. It is difficult enough to find reports of actual tests relating to the efficiency of belt drives, but there is even less in print about efficiencies of rope drive. It was in the course of a search for such data that some investigations were discovered which form the basis of a paper[1] presented at the December meeting of the Society in 1910. These tests included a large number on rope transmission for which the efficiency was carefully determined.

The description of the testing machine of that paper may be summarized here in the statement that the machine is horizontal, and

[1]A New Theory of Belt Driving, The Journal, Am. Soc. M. E., December 1910.

very compact because both driving and driven shafts are located in the measuring and straining heads respectively, and electric motors for supplying and absorbing energy are mounted directly on the shafts. The sheaves were specially designed for these tests, had diameters 2500 mm. (98.5 in.), 1500 mm. (59 in.) and 1040 mm. (41 in.), with 45 deg. V-grooves. Pains were taken to have all grooves nearly the same diameter. Most of the tests were conducted on a round manila three-strand rope 50 mm. (2 in.) diameter, 20.5 m. (67 ft.) long, the rest being taken with six-strand, so-called trapezoidal rope, with a sectional area approximately equal to that of the round rope. The majority of observations were made on transmission on the English system; the arrangement with endless rope was probably not entirely representative of the American system, the automatic adjustment of the tension being lacking.

The readings for a test included volts and amperes for power both supplied and delivered, rope-speed, rope-pull and revolutions. In order to determine the efficiency, however, only the electrical readings are required. All electrical, bearing friction, and windage losses of the testing machine were determined and allowed for so that the efficiencies reported apply to the rope only.

Upon this basis the English system showed higher efficiency than the American. All test results are reported in the form of curve-sheets, the efficiency curves of single tests being superposed, and, so to speak, enveloping curves constructed from them, from which the upper and lower limiting efficiencies at any given load or speed may be taken. On the English system with a 2-in. round rope, the highest efficiency reported is 0.89 to 0.945 at 38 kg. (84 lb.) effective pull per rope for four ropes, and 0.945 to 0.97 at 100 kg. (220 lb.) for a single rope. Pulleys of like diameters were used, while the distance between centers was about 6 m. (20 ft.). The tight part of the ropes was on the bottom. It was also observed that the use of driving sheaves smaller than the driven one did not affect the efficiency, but when the larger sheave drove, the efficiency was reduced. Furthermore, the efficiency decreases with increasing speed, one set of values being 0.95 at 13 m. per sec (2550 ft. per min.), 0.925 at 30 m. per sec. (5900 ft. per min.) and 0.90 at 39 m. per sec. (7700 ft. per min.). With an endless rope (American system) maximum efficiencies as follows were reported: 0.86 to 0.895 at 55 kg. (121 lb.) effective pull per rope. A particular value with three wraps of rope on pulleys 5933 mm. (19.5 ft.) apart, 1040 mm. (41 in.) (driver) and 2500 mm.

(98.5 in.) (driver) at a rope speed of 26 m. per sec. (5100 ft. per min.) is a maximum efficiency of 0.89 at the effective pull 50 kg. (110 lb.) per rope. Practically equal values were obtained (also equal carrying capacities) whether the tight part of the rope was on the top or bottom. The initial rope tension was also reported, the most favorable value for the 2-in. rope from the standpoint of efficiency and smooth running being about 150 kg. (330 lb.). Other interesting observations are the recommendation of a maximum rope speed of approximately 25 m. per sec. (5000 ft. per min.) and the confirmation of the prevalent opinion that with constant initial tension and rope speed the efficiency increases with the sheave diameter.

Although no figures relating to the trapezoidal rope are given, it is reported that its efficiency is lower than that of the round rope.

The formulae for calculation of efficiency developed in the previous paper were applied to several of the observations reported in the paper, but with a disappointing result, because in all cases the observed losses were much greater than calculation, not taking into account windage losses of the driven sheave and brake pulley, indicated.

The Author. Answering Mr. Stott's question as to reliability, there is very little liability of accident from rope drives if they are properly installed, and given even a fraction of the attention that other means of transmission require.

It is common practice by those most experienced in the installation of rope drives to safeguard them by placing an electrical device, called a tell-tale, below the rope where a bend is made around the sheave that will, in case the rope becomes stranded at all, make an electrical connection and either throw out a clutch stopping the drive entirely or ring a gong to notify the engineer so the rope may have immediate attention.

If a rope is properly inspected occasionally, it can be repaired at night should any distress appear, and never occasion any delay during working hours.

With either English or American methods the freedom from delays depends on the care in inspecting. There is but one rope to watch in the American system while there are many in the English. It is a common occurrence for American drives to run three or four years without touching the rope at all when properly installed.

Mr. Van Derhoef raises a hard question: Why is a rope more

efficient at low than at high speeds? My tests have not gone far enough to determine these causes. All the tests made, however, covering this long period of time, indicated in every case that the higher the speed the lower was the efficiency. It would take a long series of tests to develop why this is, but there is no doubt of the fact.

If Mr. Kenerson will refer to the various curves presented in the paper, he will see the same general results as he refers to so far as the decrease of efficiency as the speed increases. This was a general fact shown by a great many tests, and does not depend on an isolated test (tests were not made below 2500 ft. rope travel per minute).

The higher efficiency at low speeds is probably not due entirely to one cause. Skin friction and rope elasticity are possibly factors. The power to drive the sheaves and bend the rope at the high speeds may be more, proportionately, than at the slower speeds.

In Mr. Haar's discussion there are several tests reported that would not be considered good practice in rope transmission work, and therefore have little actual bearing on the subject.

A 2-in rope would not be effective on pulls of 84 lb. or even 220 lb. This would be especially true in the American system where 180 lb. in the slack side for a 1-in. rope is a common tension. The so-called American system without automatic tension adjustment is only a modification of the English system and would undoubtedly increase the friction, as it would add one sheave to the friction load without any compensating benefit. It is in no sense an American system drive, and would warrant no deductions for the American system.

It is difficult to understand why efficiencies should be lower with a large driver and a small driven sheave than with a small driver and a large driven sheave. The writer made no tests on this point, but was under the general impression there would be no difference until slipping took place, and that this would happen more quickly with the small driver than vice versa.

Mr. Haar in applying his formulae to the paper perhaps did not take into account the fact that in these tests the friction of the driven end was included with the losses in the rope itself, thus making the efficiencies much lower than they would be in the rope alone, as is usually figured.

## No. 1410

# COMPARATIVE TESTS OF THREE TYPES OF LINESHAFT BEARINGS

By Carl C. Thomas, Baltimore, Md.

Member of the Society

and

E. R. Maurer[1] and L. E. A. Kelso,[2] Madison, Wis.

Non-Members

These tests were made in order to ascertain definitely the relative and absolute amounts of power required to drive a specially constructed lineshaft carrying given loads at certain known speeds of revolution, when supported successively by three different types of shaft bearing, and to determine coefficients of friction for each type. The three types tested were ring-oiled babbitt bearings, roller bearings, and ball bearings. These are shown in Figs. 4, 5, and 6.

2 Twenty bearings of each type were used in these tests in order that representative results might be obtained. The bearings were loaned in 1909 by the manufacturers and the apparatus shown in Figs. 1, 2 and 3 was designed and built in the department of steam and gas engineering, University of Wisconsin. The design of the apparatus was accomplished with the assistance of the manufacturers of the bearings to whom preliminary drawings were submitted, and during the four years of the tests representatives of these firms have visited the laboratory for the purpose of giving whatever advice and assistance was possible. During 1910 and 1911 two theses[3] were written by senior students covering their research work on this apparatus, and Mr. William Black of the laboratory staff spent two entire summer vacations on the investigations. During the college year of 1912-1913 Mr. W. G. Moyer, a graduate student, made a very extensive set of tests which formed a decided improvement over the previous work. These tests as well as the work of the first three years were carried out under Professor Thomas's

[1]Professor of Mechanics, University of Wisconsin.

[2]Instructor in Electrical Engineering, University of Wisconsin.

[3]One by Miller and Hoyt, the other by Roth and Gaskell.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

supervision, and the results were quite satisfactorily consistent among themselves and also with reference to the preceding work. Upon the completion of these tests, however, and in order that all of the research work might be checked up independently, Prof. E. R. Maurer and Mr. L. E. A. Kelso undertook at Professor Thomas's request the work of repeating the entire investigation. This has been done within the last six months, and the results of previous investigations have not only been thoroughly checked and in some cases revised, but Professor Maurer and Mr. Kelso have been able to make numerous improvements in method and refinement of observation, and to carry the investigations farther than had been done before

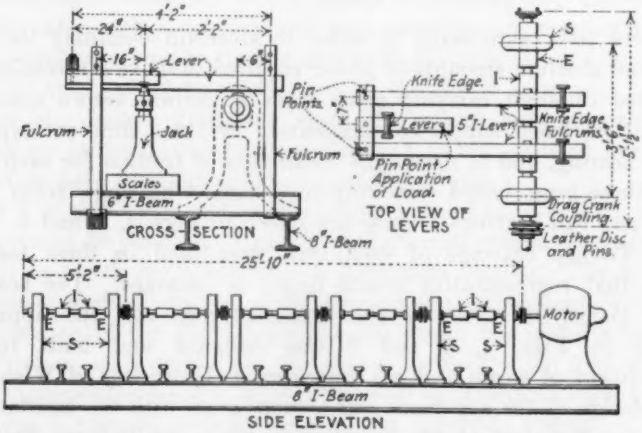



**Fig. 1 Diagram of Apparatus for Testing Lineshaft Bearings**

their work began. The following paper presents the results of their work together with parts of the work previously done. Special thanks are due to Prof. A. G. Christie and Mr. Wm. C. Rowse and to others of the laboratory staff for much valuable assistance.

3 The preliminary work, covering the first two years, showed the necessity of considering the temperature of the oil film in the babbitt and roller bearings, and it was only after careful study of the temperature question with regard to all three types, and the production of such curves as are shown on Figs. 7, 8 and 9 that satisfactory results were finally obtained. The coördinates of the curves are watts per bearing and temperature of oil, each curve representing a given load and speed. From these curves the cross curves shown on Figs. 13, 14 and 15 were obtained, the latter furnishing the data for the comparative curves on Figs. 19 and 20.

4 The principal results of the tests may be seen from the following curves:

*a* Coefficient of friction, Figs. 10, 11, 12 and 16, 17, 18
*b* Coefficient of friction, comparative results, Figs. 22, 23
*c* Power-speed curves, Figs. 13, 14, 15
*d* Power-speed curves, comparative results, Figs. 19, 20

FIG. 2 GENERAL VIEW OF APPARATUS

## DESCRIPTION OF APPARATUS

5 The apparatus consists of 25 ft. 10 in. of lineshafting in five equal sections, mounted in hangers *S* which are inverted and used as floor stands, (see Figs. 1 to 3 inclusive). The hangers are bolted to two 8-in. I-beams which are leveled upon the floor. The shafts are of cold rolled steel, 2 7/16 in. in diameter. Each section is 5 ft. 2 in. long; the adjacent sections are coupled together by means of a flexible leather disk or two straps connecting the two flange couplings. The flexible couplings prevent transmitting any part of the load applied on one shaft to either adjoining section, and also prevent binding between shafts and bearings due to possible lack of alignment.

6 A direct-current Fort Wayne motor is directly connected to one end of the shafting by means of a flexible coupling. The motor is of the interpole type with the interpoles removed, making it a shunt motor. Its rating with the interpoles is 7½ h.p., 28 amperes, 400/1600 r.p.m., 4 pole, 230 volts. The power is measured by

means of calibrated instruments, namely, voltmeter and ammeter (see Par. 15). A water rheostat is used in the field circuit to maintain a constant field, and another water rheostat is used in the armature circuit to vary the speed by varying the impressed voltage. The lost power was obtained first by taking the armature resistance for various currents and plotting a resistance, armature-current curve (see Ap-

Fig. 3 View Showing Method of Loading Bearings

pendix). From this curve was plotted an armature-$I^2R$ loss, armature-current curve. From this the armature resistance loss for any armature current was obtained. The stray-power loss was obtained by taking a stray-power loss, field current curve for each speed. Then for any speed the loss for that speed and field could be obtained. The use of the curve was better than using one point, as the points on each side of the field current formed a check. By use of the method just described, illustrated in the Appendix, the power required to run the motor alone at all speeds, without load, is accurately ascertained, as well as the power required to run the motor and shafts together, at all loads and speeds. The relative amounts of power required to overcome the friction of the various types of bearings are therefore accurately determined. In determining quantitatively the absolute amounts of power absorbed in the shaft bearings, the question arises: Is the friction in the two motor bearings the same when the motor is driving the loaded shafts at a given speed, as when the motor is run-

ning without load at that speed? In considering this matter the following line of reasoning was adopted:

*a* The friction in the motor bearings is due and proportional to the weight of the armature and to any other forces tending to add to the presence on the motor bearings.

*b* With the motor shaft and the driven shafts accurately in line, which was a condition carefully provided for

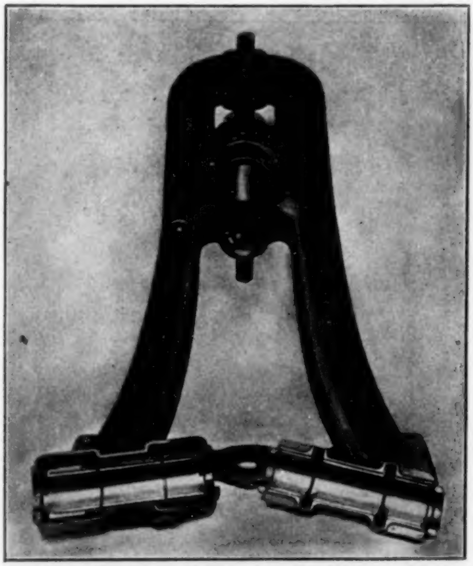


FIG. 4 TYPE OF RING-OILED BABBITT BEARING USED IN TESTS

throughout the tests, a well-made flexible leather coupling between the flanges on the motor and the driven shafts, respectively, exerts a couple on the motor shaft. Therefore any increase or decrease of load on the bearings under test, or any change of speed, changes the torque of the couple, but the couple does not affect the pressure on the bearings which remains due solely to the weight of the armature. That is, assuming accurate alignment, the armature bearing friction does not depend upon the power.

*c* If the alignment is good, even an imperfect leather coupling (the two opposite straps unequal in length) would not

seem appreciably to affect the motor friction due to weight of armature. For, suppose there were only one strap, in one-half of a revolution the pull of the strap would increase the motor bearing pressure and in the other half revolution the pull would decrease the pressure equally. Thus the friction would vary and its mean value would not be much different from that due to the weight of the armature only.

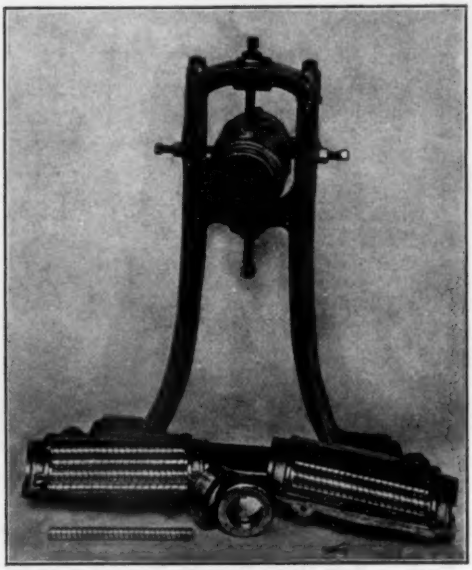


Fig. 5 Type of Roller Bearing used in Tests

*d* The total friction due to the weight of the armature is very small as compared with that due to the heavy loads carried by each of the 20 bearings. And as it is only with possible *variations* in the motor bearing friction that this part of the discussion is concerned, it appears allowable to consider the effect of such possible variations as negligible.

7 The load is applied through levers upon hardened knife edges and pin points as fulcrums. Across the top of the 8-in. I-beams and at right angles to them are bolted short 6-in. I-beams to which the fulcrums are attached. Standard 1000-lb. scales are set upon the 6-in. I-beams as shown in Figs. 1, 2 and 3.

8 A double system of leverage is used in order to get sufficient load upon the bearings with as short a length of leverage as possible. This double system of levers also serves to steady the apparatus and prevent excessive vibration. A pressure ratio of 8.33 at each bearing to one at the scale is thus realized. This was checked by an independent method of weighing the actual load resulting at the bearings from a given load on the scales.

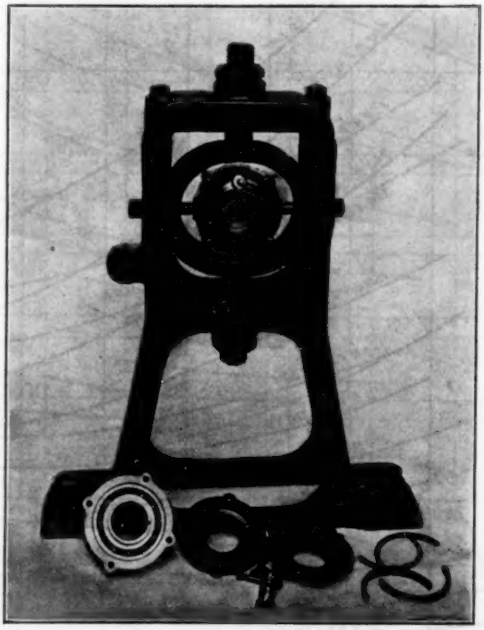


Fig. 6 Type of Ball Bearing used in Tests

9 The loads are applied to the shaft by two bearings between each pair of hangers. These bearings are identical with those in the hangers, and are supplied with knife edges which engage a V-shaped groove in the 5-in. I-beam levers. The bearings and hangers for each section are symmetrically placed with respect to the middle of the section; therefore, equal loads on the intermediate bearings *I* produce equal pressures on the end bearings *E*. These latter pressures exceed those on the intermediate bearings by one-half the weight of the shaft, or by 40 lb. This difference is regarded as negligible in the paper, and load per bearing means average of the loads

on intermediate and end bearings. The bearings are prevented from turning by short levers fastened to the bearings and resting upon uprights from the floor. Only one type of bearing was tested at a time, so that no complications might arise in establishing the losses for

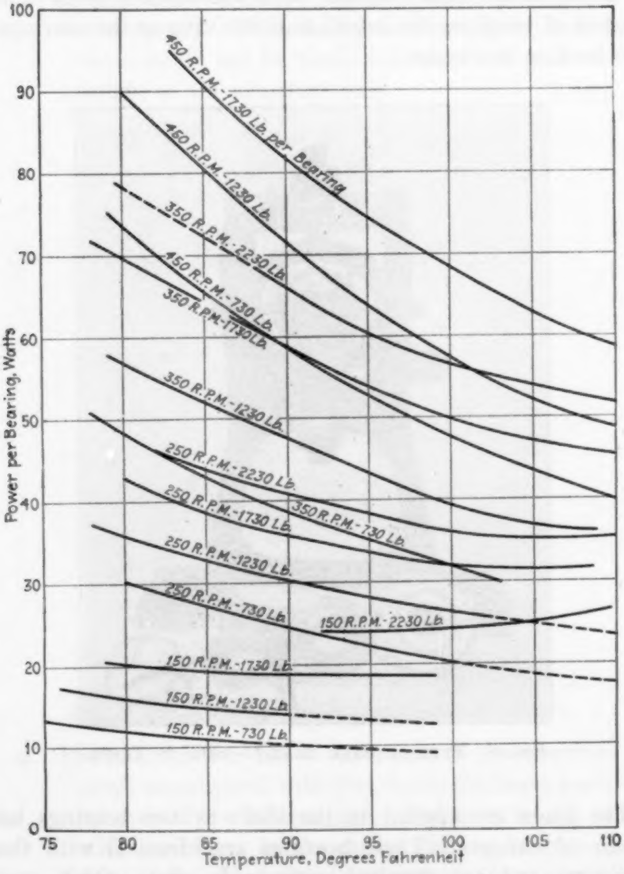



Fig. 7 Babbitt Bearings. Power-Temperature Curves for Various Speeds and Loads per Bearing

the particular bearing under test. The reason for using 20 bearings was that the amount of power necessary for a single bearing was so small as to be difficult of exact measurement. Also any single bearing might not truly represent results from that type of bearing in general.

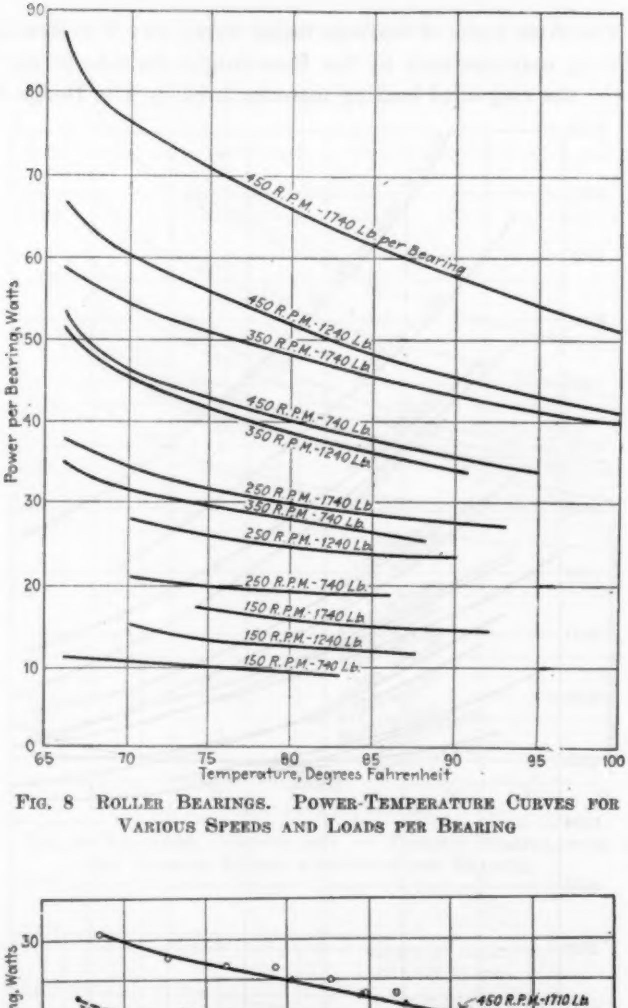



FIG. 8 ROLLER BEARINGS. POWER-TEMPERATURE CURVES FOR VARIOUS SPEEDS AND LOADS PER BEARING

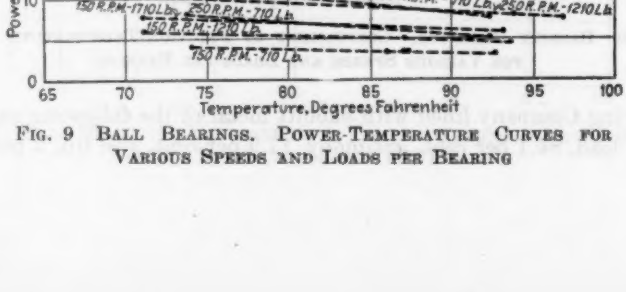



FIG. 9 BALL BEARINGS. POWER-TEMPERATURE CURVES FOR VARIOUS SPEEDS AND LOADS PER BEARING

10 The three kinds of bearings tested were: (*a*) The Hess-Bright ball bearing manufactured by the Hess-Bright Manufacturing Company; (*b*) the ring-oiled bearing manufactured by The Dodge Manu-

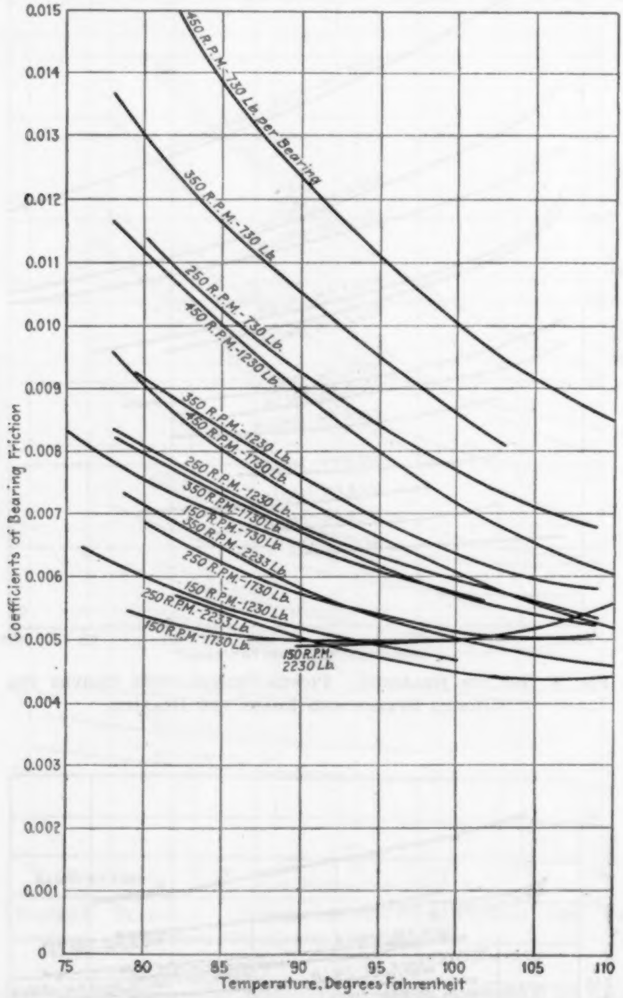



**Fig. 10 Babbitt Bearings. Coefficient of Friction-Temperature Curves for Various Speeds and Loads per Bearing**

facturing Company lined with babbitt metal of the following composition: lead, 86.1 per cent, antimony, 11.9 per cent, and tin, 2 per cent;

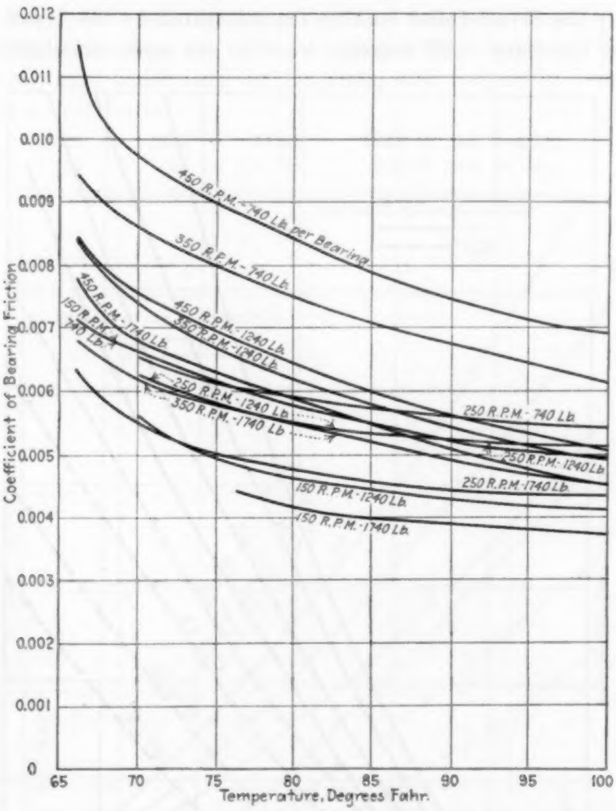



**Fig. 11 Roller Bearings. Coefficient of Friction-Temperature Curves for Various Speeds and Loads per Bearing**

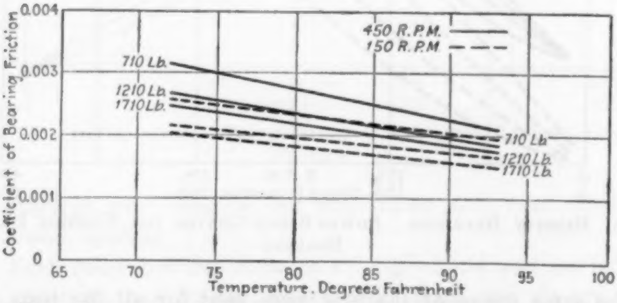



**Fig. 12 Ball Bearings. Coefficient of Friction-Temperature Curves for Various Speeds and Loads per Bearing**

and (c) the Hyatt roller bearing manufactured by the Hyatt Roller Bearing Company. All bearings were for the same size shaft and in

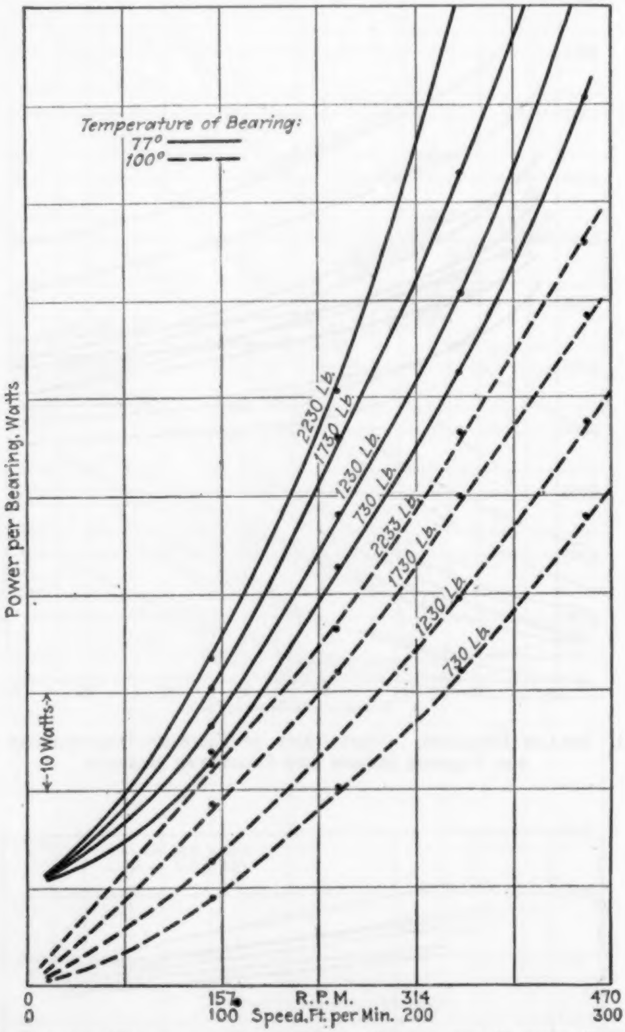



**Fig. 13 Babbitt Bearings. Power-Speed Curves for Various Loads per Bearing**

fact the same pieces of shafting were used for all the tests, except that two sections bent during the tests were replaced.

11 The babbitt bearings are 9 21/32 in. long and hence their projected area is 22.36 sq. in. The following table is appended here for converting total loads used in the tests, and frequently mentioned later, into unit loads.

| | | | | |
|---|---|---|---|---|
| 730 | 1230 | 1730 | 2230 | lb. per bearing |
| 33 | 55 | 78 | 100 | lb. per sq. in. |

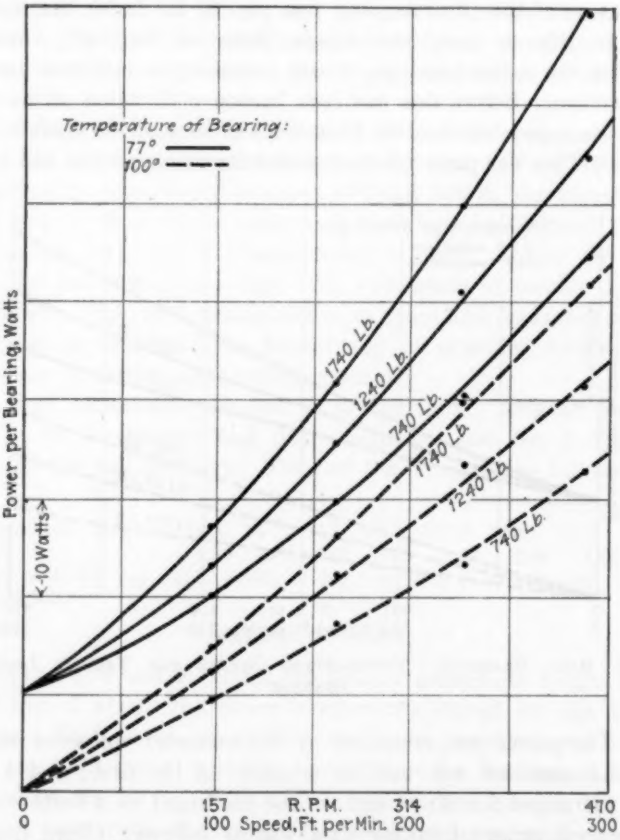



FIG. 14 ROLLER BEARINGS. POWER-SPEED CURVES FOR VARIOUS LOADS PER BEARING

These bearings were oiled by the well-known ring oiler device, two rings in each bearing.

12 Each roller bearing contains six right-hand and six left-hand rollers, 0.780 in. in diameter; six are 9 9/16 in., and six are 9 3/16 in. long. The bearings are of the type in which a cage is used for

holding one-half the rollers. Each ball bearing contains a single set of balls 9/16 in. in diameter. The diameter of the inner race through ball groove is 3.4729 in. The three types of bearing are shown in Figs. 4, 5 and 6.

13 Mercury thermometers (two in each babbitt and roller bearing, and one in each ball bearing) were used for measuring the temperature of the oil or bearing (see par. 31 for fuller description).

14 In order to avoid the endwise thrust of the shaft, when supported by the roller bearings, it was necessary to interpose two ball thrust collars. Before this was done excessive vibration of the motor and of the apparatus resulted from the tendency of the shaft to move endwise. This was particularly troublesome at high loads and speeds.

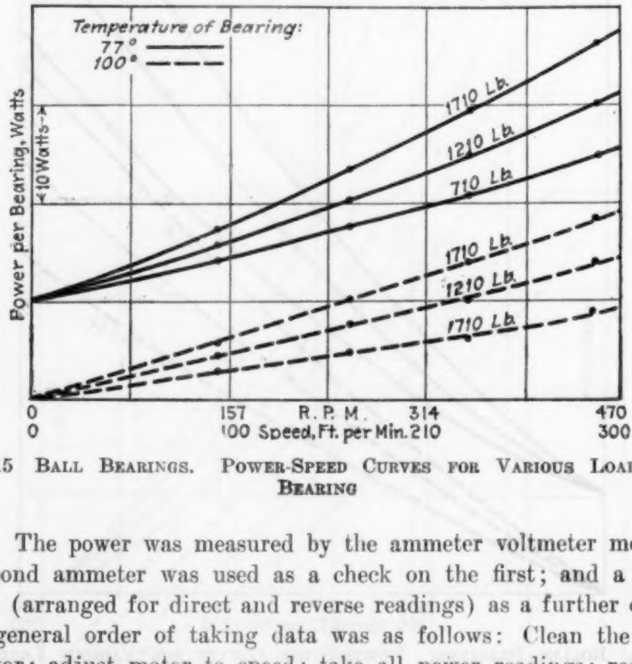



FIG. 15 BALL BEARINGS. POWER-SPEED CURVES FOR VARIOUS LOADS PER BEARING

15 The power was measured by the ammeter voltmeter method. A second ammeter was used as a check on the first; and a wattmeter (arranged for direct and reverse readings) as a further check. The general order of taking data was as follows: Clean the commutator; adjust motor to speed; take all power readings; read all thermometers; adjust speed again and repeat power measurements. This gave power both before and after temperature was taken, the mean of which would give the mean power for the mean temperature very closely. In addition the readings acted as a check on each other.

16 The manner of making a test or run was essentially as fol-

lows: Each night, the plant was run from 3 to 12 hours under the load and speed to be used in the run of the following day (but without observation), the purpose of the (preliminary) night run being to allow the shaft and bearings to adjust themselves to the conditions of the run, quite necessary as explained later. Then on the following day, the shaft was run from 3 to 6 hours and frequent observations of power and temperature were made during the run. The first few observations were made as often as practicable, about 5 minutes apart; the others, generally at 15-minute intervals, but toward the end of the run when the temperature was rising slowly, the interval was generally 30 minutes or more.

17 The speeds used in the tests were between 150 and 450 r.p.m., corresponding respectively to about 100 and 300 ft. per minute peripheral speed. Most of the loads used were between 700 and 1800 lb. per bearing, corresponding respectively to about 30 and 80 lb. per sq. in. for the babbitt bearings. All statements of results therefore are subject to the above limitations as to speed and loads, but some of the curves in certain of the figures may be extended somewhat beyond those limits for approximate results.

18 Two lubricants were used in all the tests: Atlantic Red Engine Oil in the babbitt and roller bearings, and No. 2 Keystone Grease in the ball bearings. Tests of the oil gave the following results:

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| At temperature, deg.... | 74.5 | 79 | 86.5 | 90 | 107.5 | 116 | 124 | 131 |
| Viscosity .............. | 8.86 | 7.92 | 6.06 | 5.47 | 3.73 | 2.98 | 2.66 | 2.43 |

Flash point, 400 deg.; chill point, 32 deg.; burning point, 470 deg.; specific gravity, 0.905.

## RESULTS OF TESTS

19 Figs. 7, 8 and 9 exhibit the first or immediate results of the tests. Fig. 7 shows the power-temperature curves for the babbitt bearings. Each curve shows how the power required per bearing to run the shaft varied with the temperature for the load and speed indicated on the curve. With one exception, each curve is the mean of two or more runs or curves. The solid lines represent results of continuous runs made as already explained; the broken lines are extension to "drop-down points" explained in Par. 37. The four (or in some cases three) curves for each speed have the same general characteristics. For 150 r.p.m. the power does not vary much with the temperature; at the highest load (2230 lb.) the point of minimum power for those conditions had been reached and the power was rising with rise of temperature. At the higher speed the drop in power

with increase of temperature is very decided, about 50 per cent in the entire run as represented by the upper curve, for example. The power-temperature curves for roller bearings are shown in Fig. 8. For all conditions of load and speed the power decreased with in-

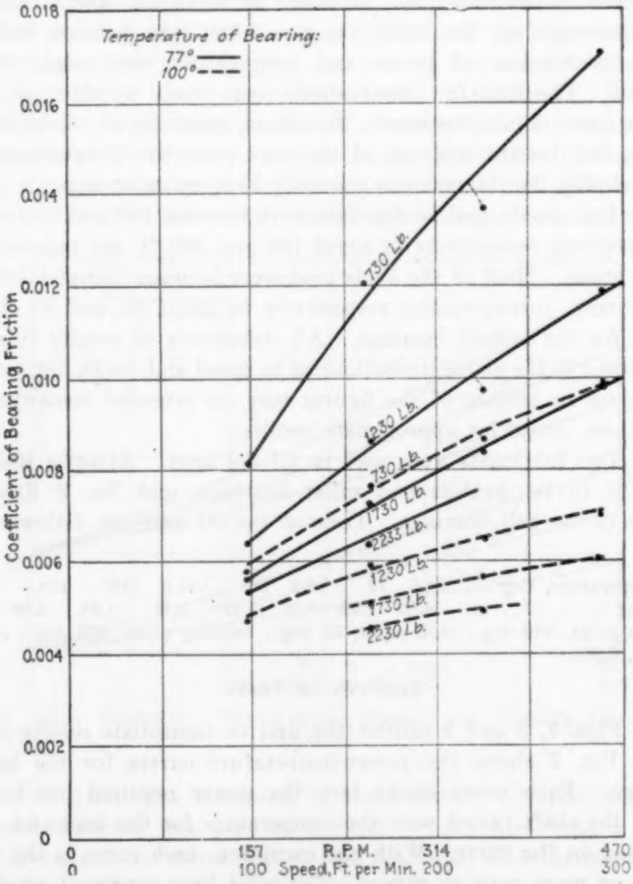



FIG. 16 BABBITT BEARINGS. COEFFICIENT OF FRICTION-SPEED CURVES FOR VARIOUS LOADS PER BEARING

crease of temperature, but the rate of decrease at the highest load and speed was not so marked as in the case of the babbitt bearings. The power-temperature curves for the ball bearings are shown in Fig. 9. At the lower speeds the power does not vary at all with the temperature, practically speaking. Before the ball bearing tests were

begun, the shaft was run in the bearings for about 100 hours, in order to settle it down to as good running conditions as had been obtained in the earlier tests. The upper curve in Fig. 9 represents the mean of two runs indicated by the circles and triangles respectively. The other solid line (for 450 r.p.m. and 710 lb. load) also is the mean

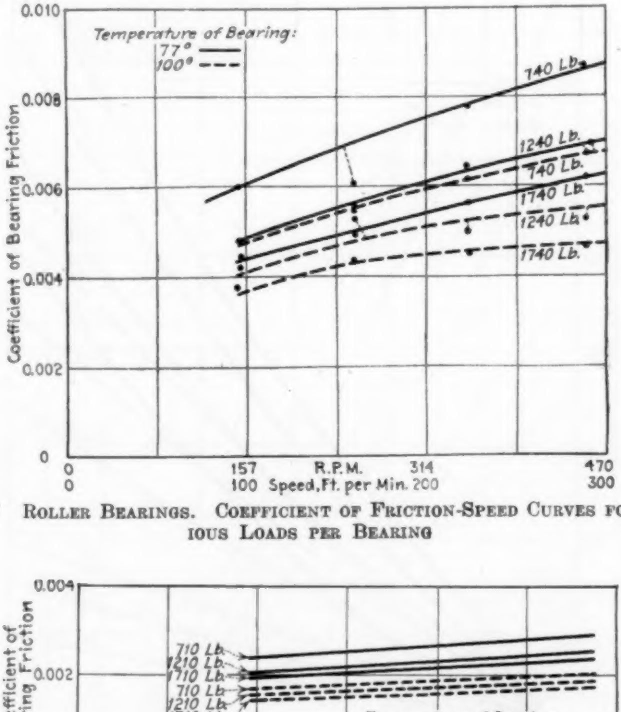



FIG. 17 ROLLER BEARINGS. COEFFICIENT OF FRICTION-SPEED CURVES FOR VARIOUS LOADS PER BEARING

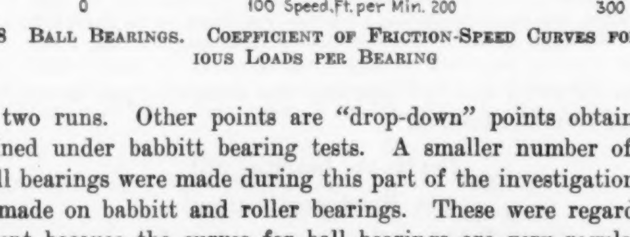



FIG. 18 BALL BEARINGS. COEFFICIENT OF FRICTION-SPEED CURVES FOR VARIOUS LOADS PER BEARING

from two runs. Other points are "drop-down" points obtained as explained under babbitt bearing tests. A smaller number of runs on ball bearings were made during this part of the investigation than were made on babbitt and roller bearings. These were regarded as sufficient because the curves for ball bearings are very regular and nearly straight, and because these results on ball bearings are in close

agreement with the extensive tests made on the same apparatus and bearings by Messrs. Black, Moyer and others.

20 Figs. 10, 11 and 12 show the coefficient of friction-temperature curves for the three kinds of bearings respectively. The co-

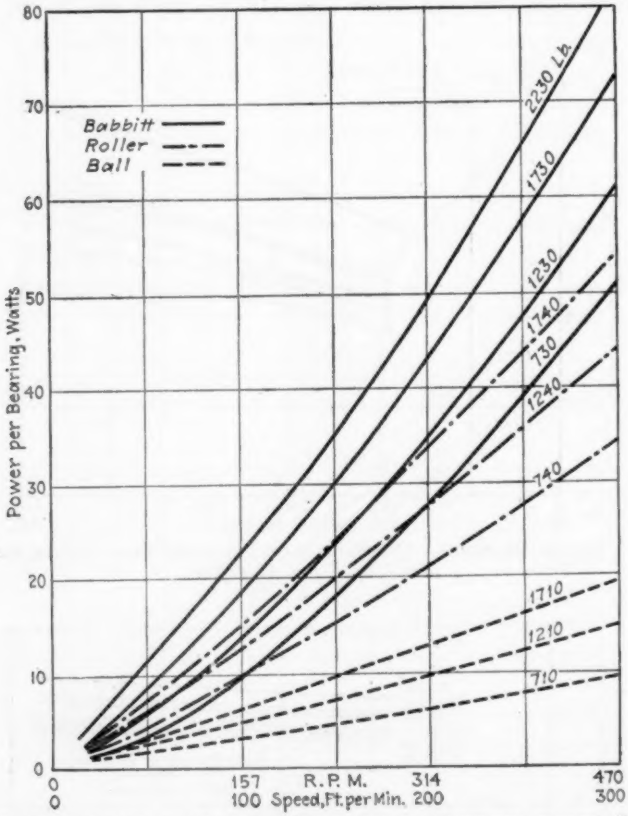



FIG. 19 COMPARISON OF POWERS CONSUMED BY FRICTION IN THE BABBITT, ROLLER AND BALL BEARINGS, AT LOADS PER BEARING AS INDICATED AND VARIOUS SPEED; TEMPERATURE OF BEARING, 100 DEG. FAHR.

efficient for roller and ball bearings was computed just as for babbitt bearings from the formula

$$f = \frac{\text{(watts per bearing) } 531}{\pi dn \text{ (load per bearing)}}$$

where $f$ is the coefficient, $d$ the diameter of shaft in inches, and $n$ is the number of revolutions per minute, 1 watt being equal to 44.26 x 12

or 531 in-lb. per minute. In Fig. 8 the curves for the intermediate speeds (250 and 350 r.p.m.) were omitted because of lack of space.

21 Fig. 13 shows power-speed curves for the babbitt bearings

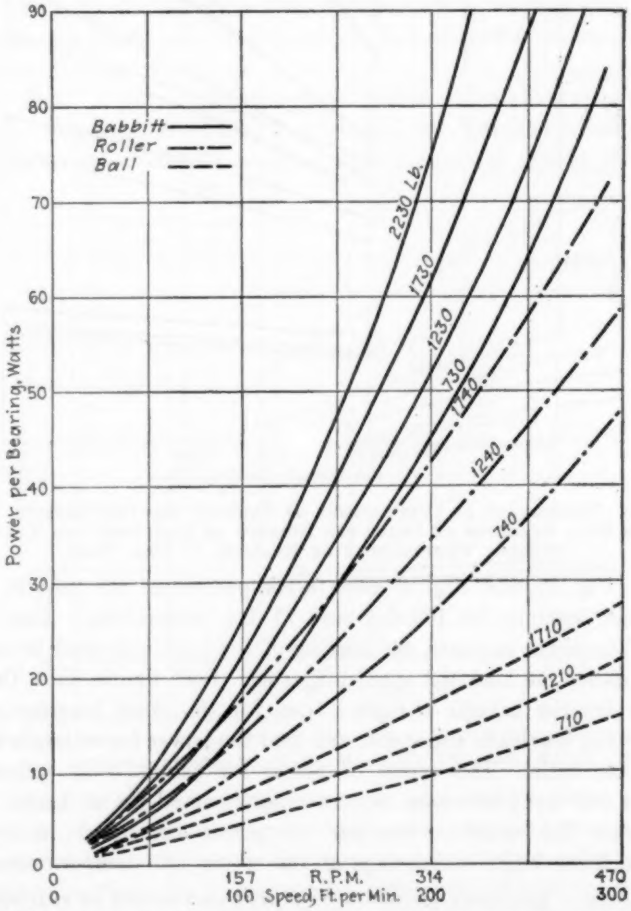



FIG. 20 COMPARISON OF POWERS CONSUMED BY FRICTION IN THE BABBITT ROLLER AND BALL BEARINGS, AT LOADS PER BEARING AS INDICATED AND VARIOUS SPEEDS; TEMPERATURE OF BEARING, 77 DEG. FAHR.

at 77 deg. and 100 deg. temperature; Fig. 14 the same for roller bearings; and Fig. 15 for ball bearings.

22 Fig. 16 shows the coefficient of friction-speed curves for babbitt bearings at 77 deg. and 100 deg. temperature; Fig. 17 those for roller bearings; and Fig. 18 for ball bearings.

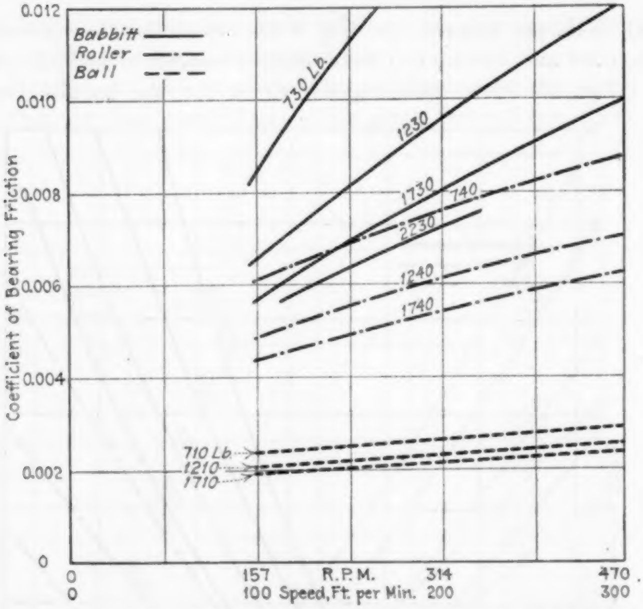



**Fig. 21 Comparison of Coefficients of Friction for the Babbitt, Roller and Ball Bearings at Loads per Bearing as Indicated and Various Speeds; Temperature of Bearing, 77 Deg. Fahr.**

23 Fig. 19 and Fig. 20 show a comparison of the babbitt, roller and ball bearings for 100 deg. and 77 deg. respectively. Each curve gives the power required per bearing of the kind indicated to run the shaft under the load and speed indicated. Both figures show that the power for the babbitt is higher than for the other bearings except perhaps at low loads and speed, and that the power for rollers is higher than for balls. The excess of power for babbitt over rollers and rollers over balls increases with increase of speed for all loads. Table 1 shows the relative amounts of power consumed in friction by the three kinds of bearings at the speeds and temperatures indi-

TABLE 1 RELATIVE AMOUNTS OF POWER CONSUMED IN FRICTION

| Bearings | 100 Ft. per Min. | | 300 Ft. per Min. | |
|---|---|---|---|---|
| | 77 Deg. | 100 Deg. | 77 Deg. | 100 Deg. |
| Ball | 1 | 1 | 1 | 1 |
| Roller | 2.2 | 2.5 | 2.7 | 3 |
| Babbitt | 3 | 3.6 | 4.5 | 4 |

cated; the relative numbers are based in each case on the average power for three loads: 710, 1210, and 1710 lb. for balls; 740, 1240, and 1740 for rollers; and 730, 1230, and 1730 for babbitt.

24 Figs. 21 and 22 present the comparison in another way. The ordinates are coefficient of friction instead of power as in the two preceding figures.

25 Figs. 23, 24 and 25 are typical curves which show respectively how the coefficients for babbitt, roller and ball bearings change with the time after starting. Although the loads and speeds were the

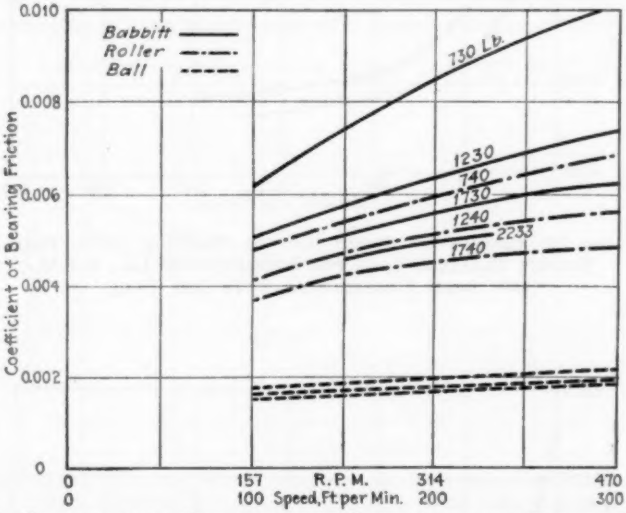



FIG. 22 COMPARISON OF COEFFICIENTS OF FRICTION FOR THE BABBITT, ROLLER AND BALL BEARINGS AT LOADS PER BEARING AS INDICATED AND VARIOUS SPEEDS; TEMPERATURE OF BEARING, 100 DEG. FAHR.

same for all kinds of bearings the room temperatures were not the same and so the curves are not comparable. In each case the coefficient and hence the power decreases rapidly at first. For babbitt and roller bearings there is a marked difference between the average coefficient for the first hour and for the second hour. Thus in the run represented by the lower curve in Fig. 23 the average coefficient for the first hour was about 0.0058 and for the second hour about 0.0048. Hence the energy wasted in friction in the first hour was about 20 per cent more than in the second hour. This effect is less marked with roller bearings.

26 Fig. 26 shows a comparison on rise of temperature for the three kinds of bearings. Curves $A$ and $B$ were obtained from simul-

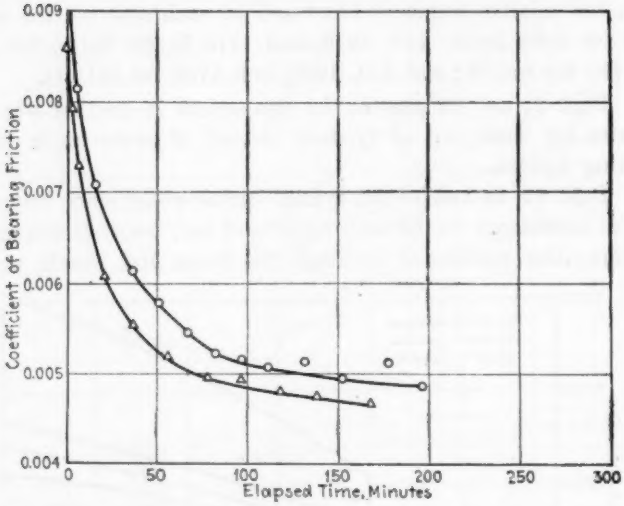



FIG. 23 CHANGE OF COEFFICIENT OF FRICTION WITH TIME. BABBITT BEARING; LOAD PER BEARING, 1730 LB.; R.P.M., 450; ROOM TEMPERATURE, 70-74 DEG. FAHR.

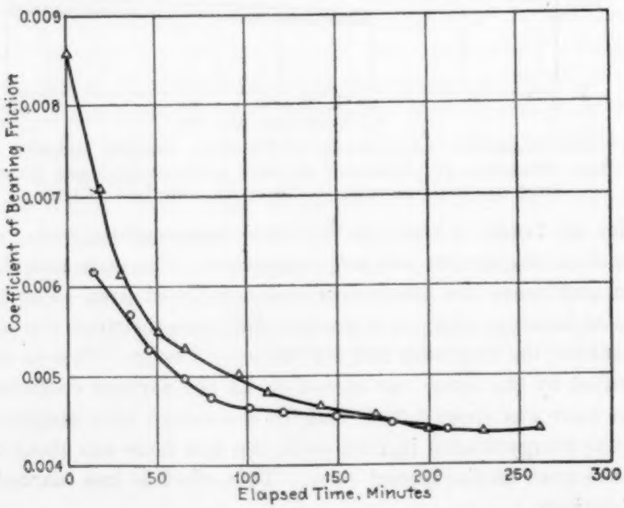



FIG. 24 CHANGE OF COEFFICIENT OF FRICTION WITH TIME. ROLLER BEARING; LOAD PER BEARING, 1740 LB.; R.P.M., 450; ROOM TEMPERATURE, 64-68 DEG. FAHR.

taneous tests on 8 babbitt and 12 roller bearings. The room conditions therefore as well as load and speed were identical for both kinds of bearings. Curves *C* and *D* were obtained from simultaneous tests on 12 roller and 8 ball bearings respectively.

## BREAKDOWN TESTS

27 In order to observe the performance of the bearings under extraordinarily heavy loads, breakdown tests were run on each type of bearing with only one section of shafting on which were four bearings. This small number of bearings was used because it was impracticable to keep close watch of a larger number of bearings and

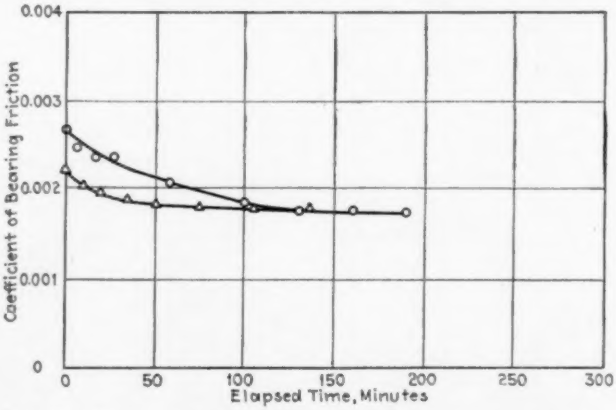



FIG. 25 CHANGE OF COEFFICIENT OF FRICTION WITH TIME. BALL BEARINGS; LOAD PER BEARING, 1710 LB.; R.P.M., 450; ROOM TEMPERATURE, 69-74 DEG. FAHR.

avoid trouble during the excessively severe conditions. The maximum load was 600 lb. on the scales, or about 5000 lb. per bearing.

28 The results are shown in Fig. 27. The speed of 200 r.p.m. was chosen because it represents about the average lineshaft speed in practice. These tests began at about 3200 lb. per bearing. Failure occurred at about 4250 lb. per bearing in the case of the babbitt, 4650 lb. in the case of the ball bearings, and about 5100 lb. in the case of the roller bearings. The quality and amount of lubricant used undoubtedly have an important effect upon the load that will cause a given bearing to fail.

29 The bearings did not in any case "fail" structurally, as the power was cut off soon after distress was manifested, but the failure was simply that of the lubricant. Breaking down of the lubricant

results in an immediate increase of the power required to maintain the original speed of rotation of the shaft in the bearings.

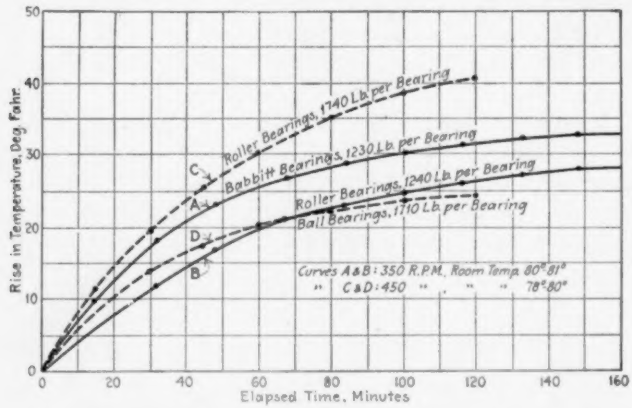



**Fig. 26 Rise in Temperature. Different Kinds of Bearings under Practically Identical Conditions**

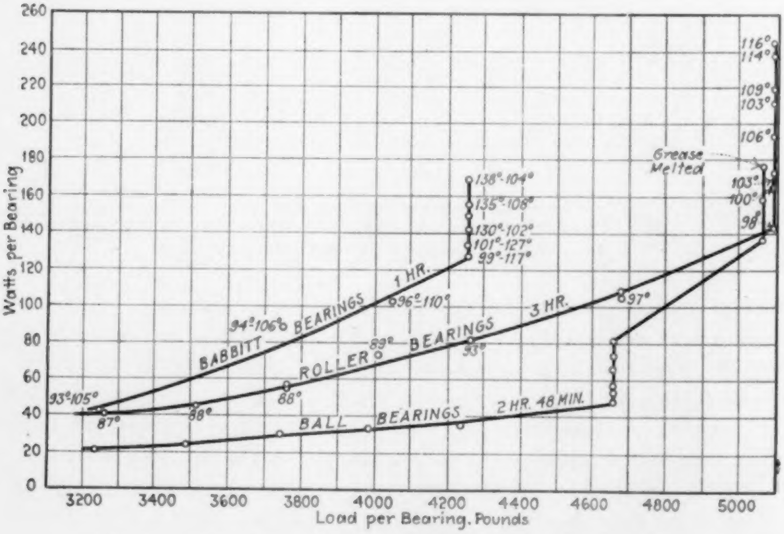



**Fig. 27 Break Down Tests, 200 R.P.M., 128 Ft. per Min.**

In each case probably only one of the four bearings used in the breakdown tests showed distress at any one time. In the case of ball bearings, this was evident as distress was manifested by disin-

tegration of the grease which "melted" and ran out of the bearing. This was accompanied by the immediate increase in power requirement, as shown in Fig. 27. The temperatures given (99 deg.-117 deg., etc.) indicate the averages for the four bearings being tested. Similar

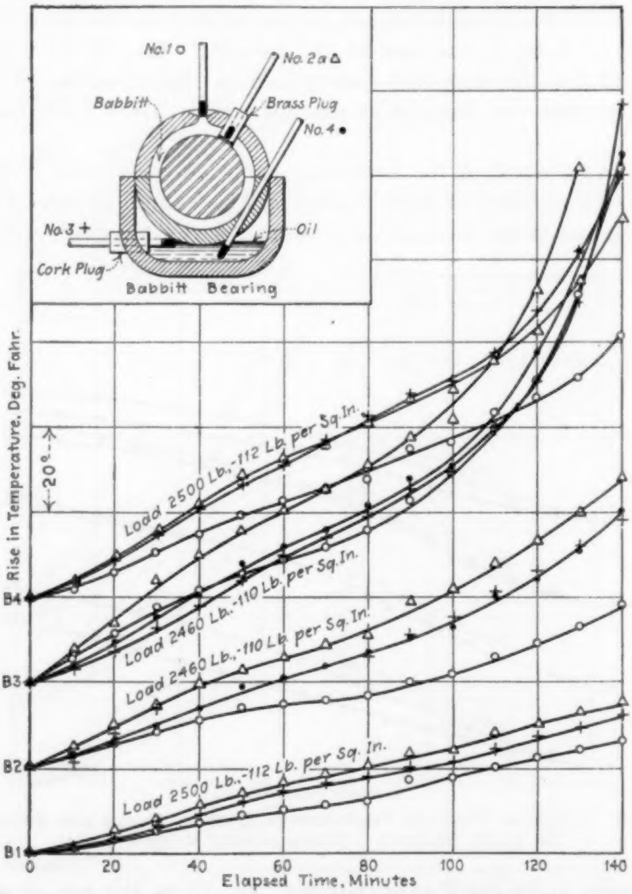



FIG. 28 FIRST COMPARISON OF FOUR METHODS FOR DETERMINING TEMPERATURE. SPEED, 260 R.P.M.; ROOM TEMPERATURE, 89 DEG. FAHR.

behavior on the part of the babbitt and the roller bearings indicated that at least one of the four under test was suffering from an approach to "metal-to-metal" contact. The bearings were not injured by these endurance tests, and all were used in subsequent tests at the more usual speeds and pressures.

DEVELOPMENT OF METHODS OF TESTING

30 The following account is given in order to describe the difficulties as they came up and the means of overcoming them. In all the runs made earlier, the temperature had been measured by placing the thermometer either on the babbitt in the sprue hole or in the oil hole, in the case of the babbitts. In case of the roller bearings the thermometers were placed in the oil holes. The ball bearings were not thought to rise appreciably above room temperature.

31 As a result of the knowledge gained in those runs it was believed the methods of measurement did not afford a very close approximation to the oil film temperature. A run was made with ther-

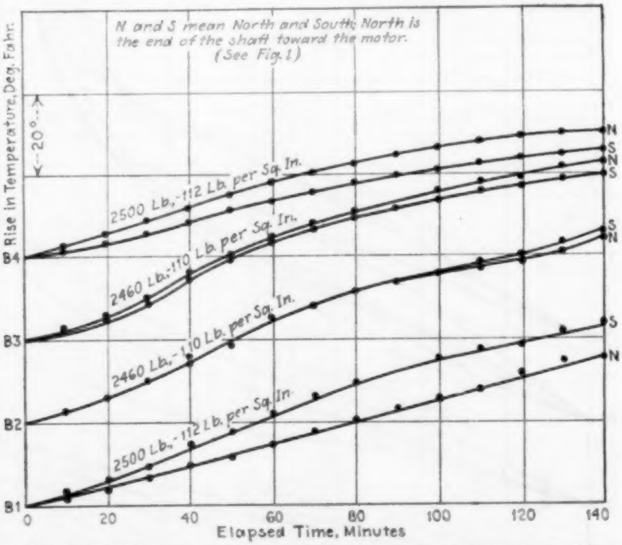



FIG. 29 CURVES TO COMPARE TEMPERATURE RISES IN NORTH AND SOUTH ENDS OF EACH BEARING. THERMOMETER METHOD NO. 4

mometers as in early runs, position No. 1; in the top in a brass casing, so as to be wet by the oil from the oil rings No. 2; in the side of the casings No. 3; and in the oil well by the side of the rings as No. 4. The results are shown in Fig. 28. Position No. 4 was used in the next tests as it was believed that since the oil rings were continually carrying up an excess of oil which generally flowed back into the oil well, it would give a close approximation to the average bearing temperature.

32 The next question was, do the bearings heat alike? That is, will the two center bearings heat alike, and the two end bearings heat alike, and do the two ends of any bearing heat the same? Data for Fig. 29 were taken on one section of the shaft (four bearings), the

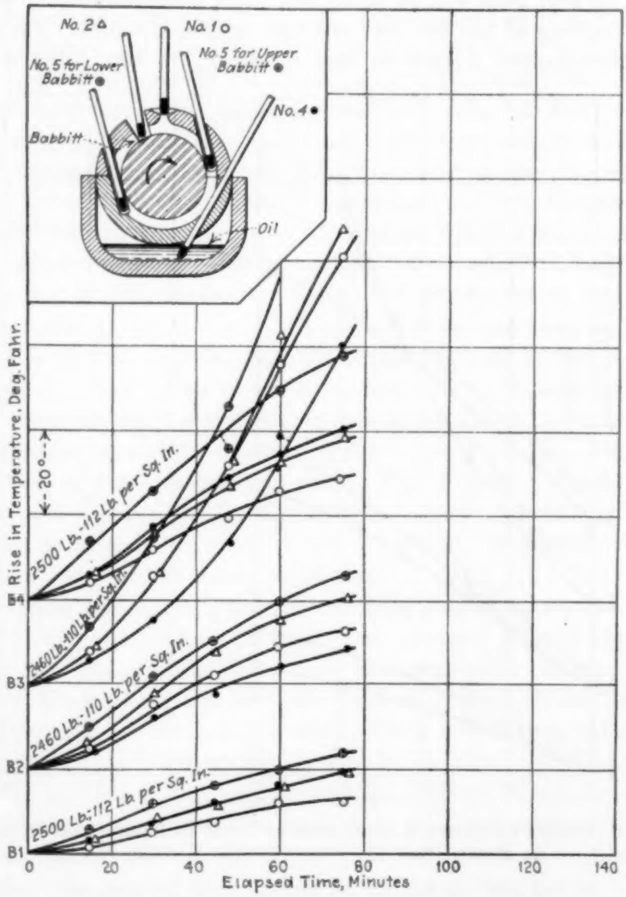



FIG. 30 SECOND COMPARISON OF FOUR METHODS FOR DETERMINING TEMPERATURE. SPEED, 260 R.P.M.; ROOM TEMPERATURE, 74 DEG. FAHR.

thermometers being as in method 4. The results indicated that not only do the end and center bearings not heat alike, but the two ends of the end bearings heat unevenly. It was now decided to use two thermometers on each bearing, placed as in method 4, those in the end

bearings being on opposite sides of the shaft from those in the middle bearings, and on the side which the loaded surface of the shaft in the bearing was approaching in order to get the temperature of the oil as soon as possible after it left the bearing.

33 In Fig. 28 it will be noted that there is considerable difference in the heating of the two end and two middle bearings. They were next interchanged $B_4$ for $B_1$ and $B_3$ for $B_2$ to see if the unequal

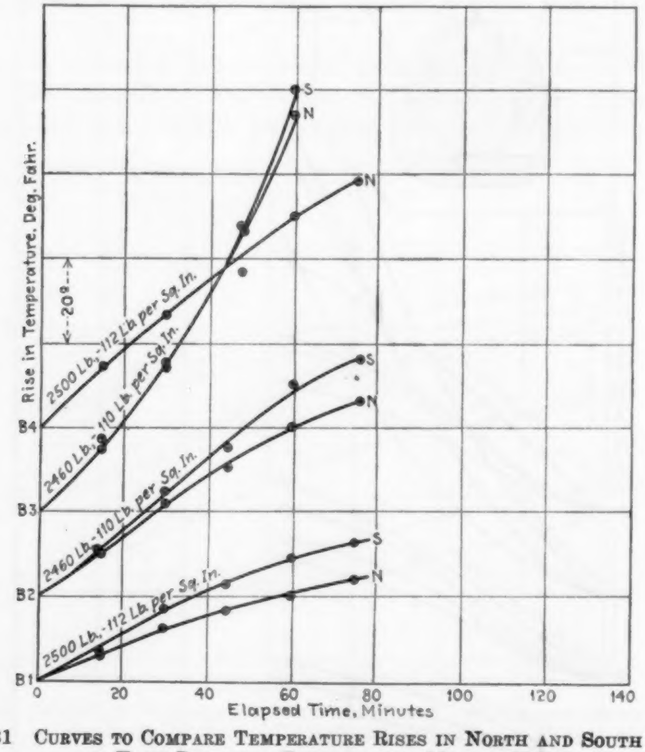



FIG. 31 CURVES TO COMPARE TEMPERATURE RISES IN NORTH AND SOUTH ENDS OF EACH BEARING. THERMOMETER METHOD No. 5

heating could be eliminated. These results, not given, still indicated an unequal heating.

34 It was found that some of the bearings got much warmer than the average. Copious oiling of these was tried but was not effective. Electric fans were next tried; these were able to keep the bearings cool enough so that the temperature of the individual bearings was not far from the average.

35 It was felt that while method 4 gave the actual temperature of the oil in the oil well, it still did not give a very close approximation to the oil film temperature. Having read reports[1] of Stribeck's experiments along the same line, it was decided to try thermometers inserted in the babbitt itself. A run was made (Fig. 30) to compare this method with the other methods. From this run it was decided to use method 5. A run (Fig. 31) was next made with thermometers as in method 5, to see if the two ends of any bearing heated alike. The results of this run showed the necessity of using two thermometers per bearing. The tests were then made with thermometers, method 5, two thermometers to each bearing. The temperature recorded was the average of the readings from 40 thermometers. The thermometers were calibrated but the error of the average was less than the error of a single observation. The runs proper were made with fans to cool any bearing getting excessively warm, and thermometers, method 5.

36 Curves 1*a*, 1*b*, 1*c* (Fig. 37) indicated that the bearings should run for a time at the load and speed they were to be tested at, in order to get them down to an adjustment. The remainder of the curves were run with a preliminary run of from two to three hours. The general procedure was to make a run at any load and speed, one in the forenoon and one in the afternoon; then the apparatus was adjusted for the next load and speed and allowed to run from two to three hours, when it was shut down. The graphical logs show how near the two runs checked.

37 There were quite a number of "drop points" obtained. These were obtained by dropping on either load or speed, thereby obtaining a point of a higher temperature than was obtained in the run. That is, at the lower speeds and loads the bearings did not attain the comparatively high temperatures to which it was desirable to extend the curves for low loads and speeds, as well as for high. Therefore after high load and speed runs had warmed the bearings to a desired temperature, the load or speed, or both, were at times reduced in order to obtain power readings for those loads and speeds at higher temperatures. In general these points lie on the curves, but in no case did they check so closely as to warrant not making a run at that load and speed, the drop points acting more or less as a check, and not being taken as final results.

38 Since in the babbitt bearing it was found the early used method did not approximate very closely the actual film temperature,

[1]Zeitschrift des Vereines deutscher Ingenieure, vol. 46, p. 1341.

and that the heating was unequal, similar tests were made with the roller bearings. As a result of these tests it was decided to use two thermometers on each bearing, placing them so the bulbs were as close as possible to the point on the steel lining where the load was applied. In addition, a small hole was drilled so the oil would be forced around the bulb.

39 Wearing-down runs on the roller bearings were made of about the same lengths as on the babbitts until test 3*a* was run (Fig. 39). As will be noticed, there is quite a hump in the curve. The cause of this is not absolutely known. At about the time the hump appeared, it was noticed that some of the bearings were getting dry. They were oiled; then after another reading the bearings were all oiled until oil ran out the ends. Inasmuch as that was the only change observed, it was thought added oil required more energy to carry it around the shaft. It will be noticed that 3*b* is higher than the first part of 3*a*, while the lower part is lower. During this period and for a day or two later, the oil was forced out of the ends of the bearings. The load was then raised to the highest load and curves 4*a*, 4*b*, 4*c*, 4*d*, and 4*e* were run in succession (Fig. 38), after which 5*a* and 5*b* were run at same load and speed as 3*a* and 3*b*. As will be noted, 5*a* and 5*b* are quite appreciably lower than 3*a* and 3*b*, indicating a wearing down to a better adjustment. Beginning with 5*a* and 5*b*, the preliminary runs were all-night runs in order to get a better adjustment. It will be noted on curves 7, 9, 10, 11, 12 and 14 (Figs. 38 and 40) the second or *b* curve does not at first coincide with the *a* curve. This was explained by the fact that curve *a* was started at room temperature, while curve *b* was started while the shaft and bearings were warm from run *a*, making the actual film temperature of *b* relatively nearer the thermometer reading than in curve *a*. In order to see the effect of copious oiling on the power consumption, no more oil was put in the bearings until the end of the series when a curve, under same load and speed as 3*a* was run. At the end of about an hour, the bearings were copiously oiled. The curve was humped very similar to 3*a*.

40 It had been thought that the ball bearings did not heat up appreciably and that the power did not change with temperature. A run was made to determine whether they did heat up or not. The thermometers were placed in holes drilled in the casings so as to bring the bulb as close as possible to the outer ball race. The results showed that the bearings did heat to a certain extent and did not heat alike. It was also found that they needed a preliminary wearing-down run to get them to adjustment.

41 A power-temperature curve for ball bearings (Fig. 9) at highest load and speed was run and then duplicated; then drop points obtained for all other loads and speeds at the high temperature. Drop points were taken for the low or room temperature and a curve at lowest load and high speed taken and duplicated. The two full curves being nearly straight lines, it was judged all power-temperature curves would be practically straight, and were so drawn, connecting previously obtained high and low points. As a further check, other intermediate points were taken which fell on the curves as shown. The tests on the ball bearings were in all cases much more easily made than were those on the other types of bearing, because the ball bearings are not affected by such extensive temperature changes as is the case with the others. As previously stated, the ball bearing tests made in this part of the investigation check the earlier results quite exactly.

# APPENDIX

## DIAGRAMS OF MOTOR LOSSES AND GRAPHICAL LOGS OF TESTS

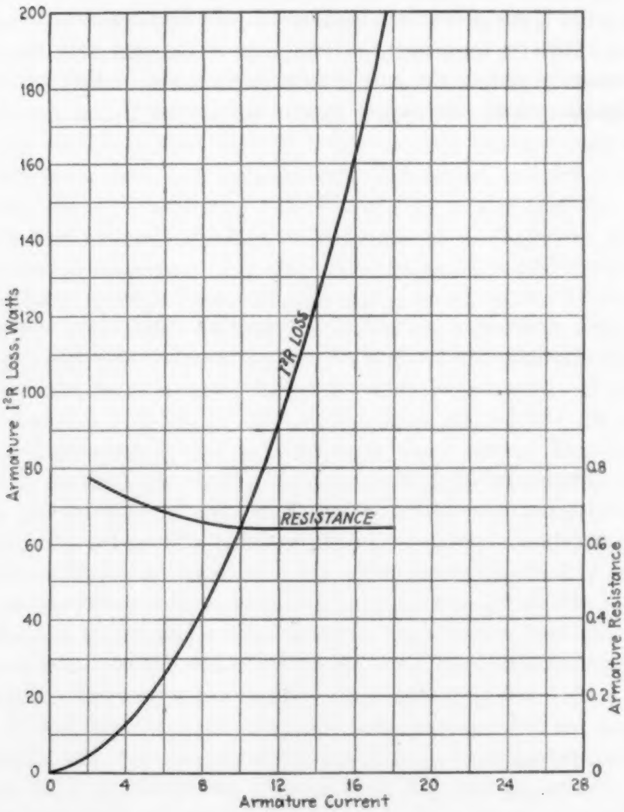



Fig. 32 Armature Resistance Loss for Motor used in Tests

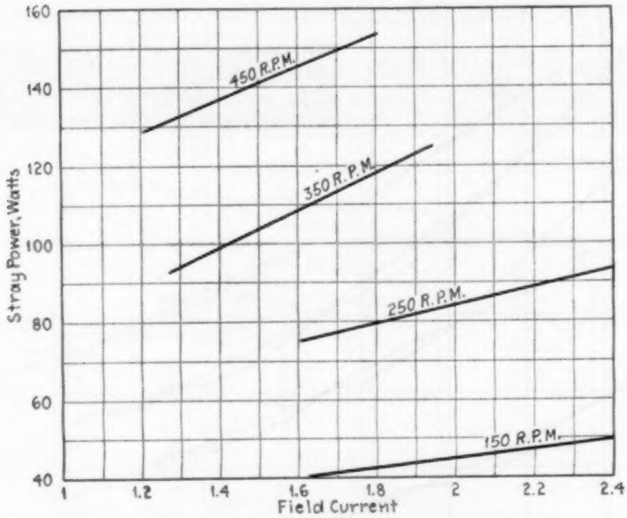
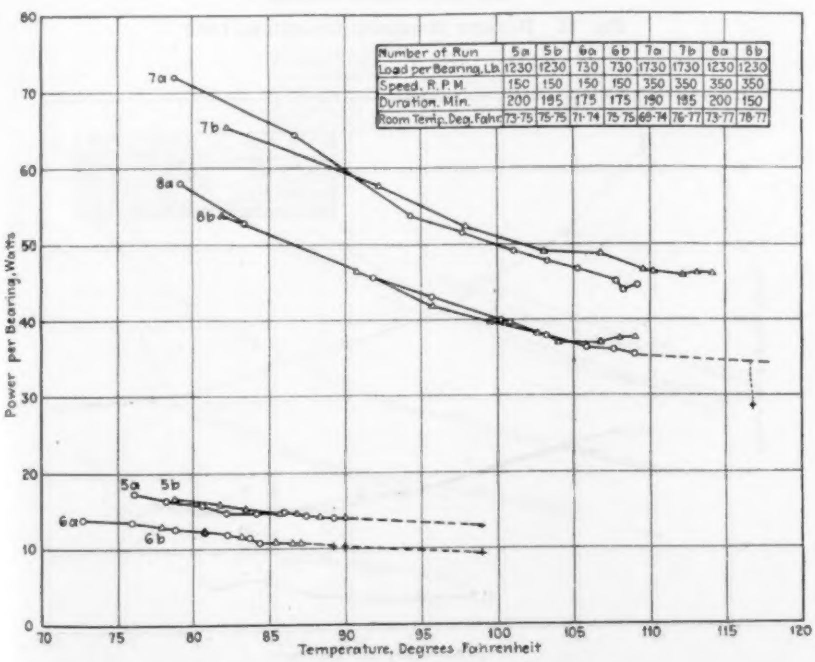



FIG. 33 STRAY POWER LOSSES FOR DIFFERENT SPEEDS AND FIELD CURRENTS




| Number of Run | 5a | 5b | 6a | 6b | 7a | 7b | 8a | 8b |
|---|---|---|---|---|---|---|---|---|
| Load per Bearing, Lb. | 1230 | 1230 | 730 | 730 | 1730 | 1730 | 1230 | 1230 |
| Speed, R.P.M. | 150 | 150 | 150 | 150 | 350 | 350 | 350 | 350 |
| Duration, Min. | 200 | 195 | 175 | 175 | 190 | 195 | 200 | 150 |
| Room Temp. Deg. Fahr. | 73-75 | 75-75 | 71-74 | 75-75 | 69-74 | 76-77 | 73-77 | 78-77 |

FIG. 34 BABBITT BEARINGS, GRAPHICAL LOGS

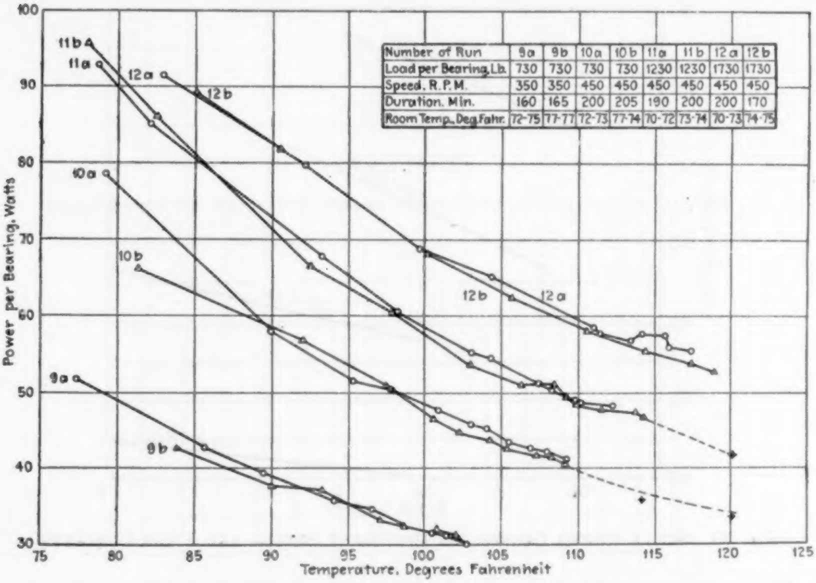



FIG. 35 BABBITT BEARINGS, GRAPHICAL LOGS


FIG. 36 BABBITT BEARINGS, GRAPHICAL LOGS

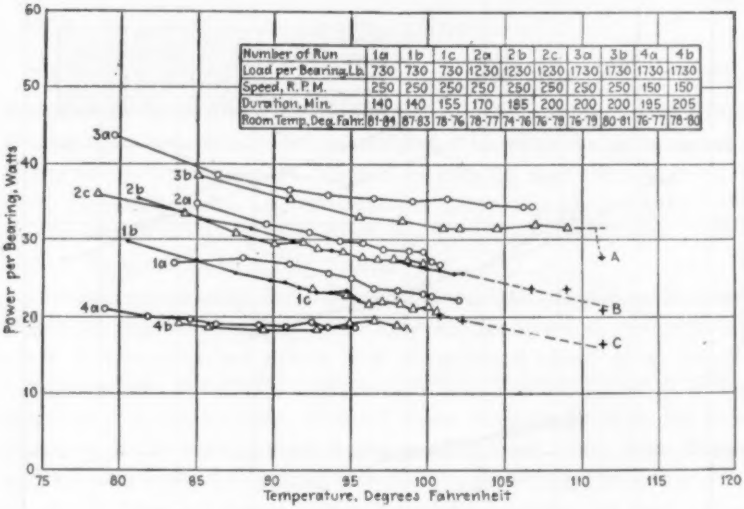



FIG. 37 BABBITS BEARINGS, GRAPHICAL LOGS

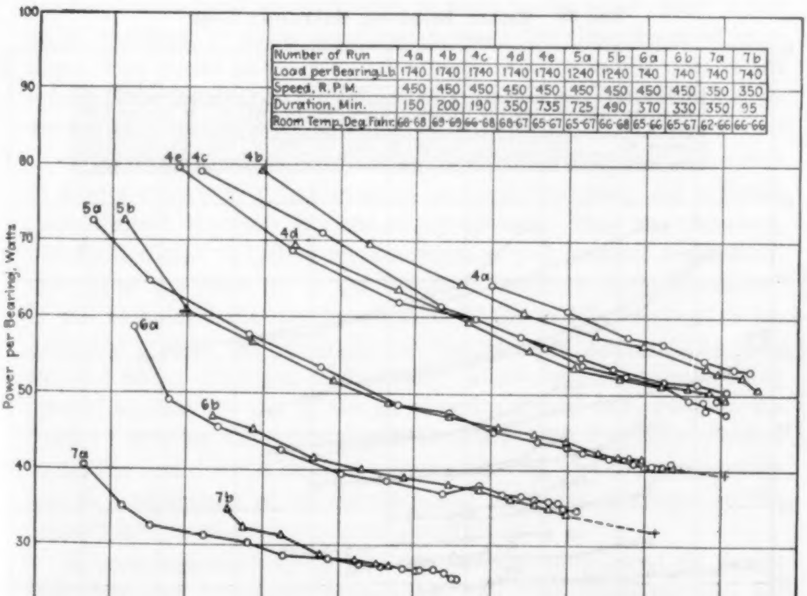
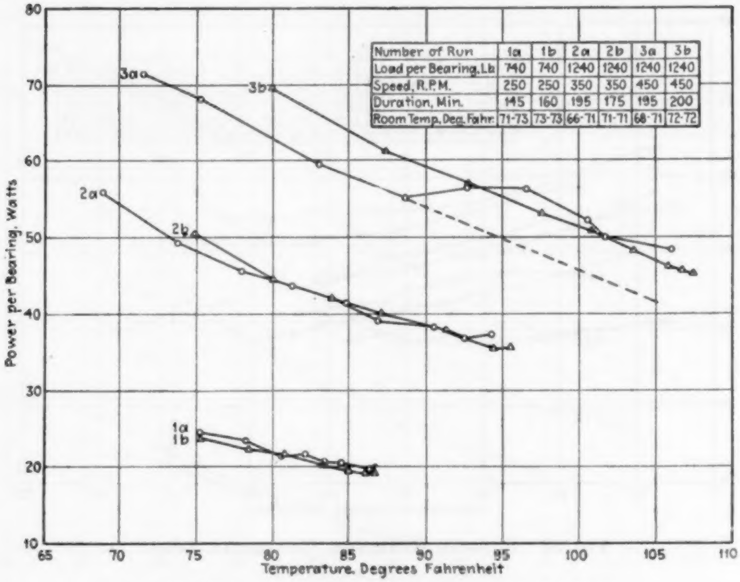
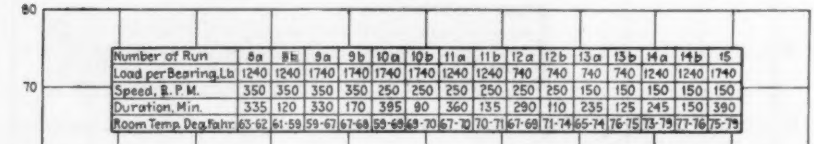



FIG. 38 ROLLER BEARINGS, GRAPHICAL LOGS




| Number of Run | 1a | 1b | 2a | 2b | 3a | 3b |
|---|---|---|---|---|---|---|
| Load per Bearing, Lb | 740 | 740 | 1240 | 1240 | 1240 | 1240 |
| Speed, R.P.M. | 250 | 250 | 350 | 350 | 450 | 450 |
| Duration, Min. | 145 | 160 | 195 | 175 | 195 | 200 |
| Room Temp. Deg. Fahr. | 71-73 | 73-73 | 66-71 | 71-71 | 68-71 | 72-72 |

Fig. 39 Roller Bearings, Graphical Logs




| Number of Run | 8a | 8b | 9a | 9b | 10a | 10b | 11a | 11b | 12a | 12b | 13a | 13b | 14a | 14b | 15 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Load per Bearing, Lb | 1240 | 1240 | 1740 | 1740 | 1740 | 1740 | 1240 | 1240 | 740 | 740 | 740 | 740 | 1240 | 1240 | 1740 |
| Speed, R. P. M. | 350 | 350 | 350 | 350 | 250 | 250 | 250 | 250 | 250 | 250 | 150 | 150 | 150 | 150 | 150 |
| Duration, Min. | 335 | 120 | 330 | 170 | 395 | 90 | 360 | 135 | 290 | 110 | 235 | 125 | 245 | 150 | 390 |
| Room Temp. Deg. Fahr. | 63-62 | 61-59 | 59-67 | 67-68 | 59-69 | 69-70 | 67-70 | 70-71 | 67-69 | 71-74 | 65-74 | 76-75 | 73-79 | 77-76 | 75-79 |

Fig. 40 Roller Bearings, Graphical Logs

# DISCUSSION

E. H. Ahara (written). This paper is of vital interest to every manufacturer, and will be a decided addition to our engineering literature on this most practical subject. It would be most desirable if our universities could be induced to take up scientifically many of these problems which are constantly confronting the manufacturers of the country who have neither the time, equipment nor staff to handle them except in a very limited way.

In connection with this subject, I have been unable to find anywhere the ratio in heating value between the speed of rotation of a shaft, and pressure per square inch of projected area. Many times a question of the limit of a bearing's ability to carry a load at a high speed of rotation becomes of much more importance than the exact efficiency of the bearing from a power standpoint. Mr. John Harper told me of a case of this kind in which by exceeding the limits of all known authorities on some large babbitted bearings, he was enabled to save nearly a quarter of a million dollars in installation charges.

I notice on the lower pressures, as given in the paper, there is but slight variation in power consumed between the three types of bearings. This would be the case in ordinary practice as the usual shaft transmission bearing of 2 7/16 in. in diameter has less than 500 lb. weight to support.

A babbitted transmission bearing usually has the pressure applied to it in a downward direction or at an angle downward and sidewise (sometimes horizontal) and but rarely upward. They are, therefore, usually designed and oiled for a downward pull, and oil is fed from the top of the liner, as in the babbitted bearings under discussion. It will be noticed that the temperature of the middle bearings of the babbitted type in the university test heated much more rapidly than the end ones. This was undoubtedly caused by the pressure being upward against the cap of the bearings instead of downward on the liner. I have no doubt that in the breakdown test Professor Thomas found the first distress came in a center bearing, and that the average rise in temperature of all the tests was markedly increased by the center bearings where heavy pressures were used.

In some tests made in 1907 by the writer at the plant of the Dodge Manufacturing Company, Mishawaka, Ind., the apparatus was arranged so that the pressure was downward on the test bearing, it being placed between two bearings which took the upward pressure and thus having but half the load of the test bearing. The test was

undertaken to determine the limit of power and speed at which these same babbitted bearings as spoken of in the paper under discussion could be run before the film of oil would no longer stay between the babbitt and the shaft. After much preliminary work a speed of 2200 r.p.m. was settled on, giving a peripheral shaft travel of 1404 ft. per min., and the pressure was gradually increased until the practical limit of the oil was reached; the temperature was measured as in method No. 4 of the paper. None of the bearings were run to destruction; the test at 2200 r.p.m. and a definite pressure being continued until there was no further rise in temperature, when a higher pressure was taken. Different grades of babbitt were used to line the bearings, the oil in all cases being Atlantic red.

Pure lead was tested up to 100 lb. per sq. in. of projected area with a temperature of 292 deg. fahr.; pure tin to 124 lb. with a temperature of 276 deg. fahr.; genuine babbitt with a composition of copper 4 parts, tin 89 parts, antimony 7 parts, reached a pressure of 124 lb. with a temperature of 282 deg. fahr.; and a composition of copper 2 parts, tin 40 parts, antimony 14 parts and lead 44 parts, reached 144 lb. with a temperature of 290 deg. fahr. A composition of tin 5 parts, antimony 17 parts and lead 78 parts, reached 83 lb. with a temperature of 298 deg. fahr., and a composition of antimony 15 parts and lead 85 parts reached only 62 lb. with a temperature of 314 deg. fahr. At low pressure no appreciable difference was noticed in the heating values, but in the high pressures the tin-based babbitts with small copper contents were found much superior. In view of these tests I think it would be wise for Professor Thomas to have the babbitt metal of the babbitt bearings analyzed and referred to in the paper.

William Kent regarded the paper as a most valuable contribution to the literature on friction, but would like to see it supplemented with results so illustrated that they would be more immediately available to engineers in designing. For example, referring to Table 1, the relative amounts of power consumed in friction, Mr. Kent would like to have seen that table extended to show what relation the amount of power consumed in friction was to the power transmitted. What designers would like to know are the limitations of load and speed for bearings of different kinds, such as ball bearings, roller bearings, and babbitt bearings, with different kinds of lubricants; in other words, the results of this investigation should be put into convenient form for designers.

Selby Haar, referring to the coefficient of friction as it changed with time of operation, asked whether Professor Thomas had any figures as to the coefficient of friction at the actual moment of starting. That was frequently quite an important thing to know in determining the torque of motors or other apparatus which drives a lineshaft and machinery connected to the shaft. He believed that the ratio of initial to final coefficient of friction was frequently much larger than was given by Professor Thomas.

The Author. replying to Mr. Kent's question, the curves in Figs. 9 to 19 give the relative and absolute amounts of power required by the various bearings, over the whole range of loads and speeds covered in the tests, and also coefficients of friction under all conditions obtained in the tests. Table 1 is not intended to give more than a few figures, taken from the curve, so as to present a general idea of the relative power consumption of the three types of bearing.

Mr. Haar's question as to change of coefficient of friction with time after starting, and its value at starting, is answered, at least in part, by the curves in Figs. 22 to 25 inclusive (shown only in the complete paper).

Regarding Mr. Ahara's question, as to the loading of the two center bearings on the cap instead of on the bottom of the bearing, the design of the babbitt bearings, with respect to lubrication, made it necessary to apply the load in the manner described. On each section of shaft, two bearings were thus loaded on the bottom and two on the cap. The extent of surface of babbitt was the same in the two cases, and the possible unequal distribution of the lubricant would seem to be the principal disadvantage involved. In the breakdown tests the two center bearings did heat more extensively than did the end bearings, and very probably for the reason mentioned by Mr. Ahara, and referred to in the paper. In the breakdown tests the bearings were loaded probably to six times the normal working load for such bearings.

There is this to be said, that in practice lineshafts are likely to be loaded from various angles, and the conditions obtaining during the tests, as to direction of pressure, might be duplicated. The writers recognized, however, that from the standpoint of effective lubrication, especially during heavy loads, it would be preferable to carry all loads on the bottoms of the babbitt bearings. All three types of bearing were loaded in the same manner as was the case with babbitt bearings. The regularity of the curves obtained in the tests, and the

results of a series of temperature investigations, in which the load per bearing was about 2500 lb., indicate that the lubrication of the bearings was quite satisfactory up to the extreme breakdown loads of from 4000 to 5000 lb. per bearing, represented in Fig. 21. It will be noticed that the unit bearing pressures employed during the regular tests described were not usually high, being between limits of about 30 and 100 lb. per sq. in. of projected area, for the babbitt bearings. During the break-down tests the loads at which the lubricant broke down, in the various types of bearing, varied from about 180 to 230 lb. per sq. in., referred to the area of babbitted bearings. The temperatures attained are shown approximately in Fig. 27.

Analysis of the babbitt had not been made up to the time of writing this paper, but will be made and reported at a later date. It is planned to extend the work so as to include other features of performance, including the question of ultimate time of endurance under the working conditions found in practice.

# No. 1411

# PITOT TUBES FOR GAS MEASUREMENT

By W. C. Rowse, New York

Junior Member of the Society

The measurement of gases is receiving increased attention in this country and in Europe. The vast quantities of natural and manufactured gas consumed for power, lighting and heating must be measured with precision not only because of their value but because modern business methods demand accuracy. Blowers and ventilating fans are usually sold under a guarantee that they will deliver a certain volume of air or gas under given conditions, but often the two different methods employed by the seller and the purchaser in measurement disagree and neither one has indisputable evidence that his method is correct. Other instances might be cited to show the need of more knowledge concerning the accuracy of various methods of measuring gases.

2 The pitot tube as a means of measuring gases has been described by many writers and investigators.[1] It has the advantage of being correct in principle, inexpensive, portable, and is in general easily applied. But the accuracy of the different forms of pitot tubes has long been questioned, the maker of each form supposing that his tube is correct, while as a matter of fact no two forms of tube agree. There has long been need for a careful, scientific study of the pitot tube for the measurement of gas and a fair comparison of the different forms in common use.

3 The purpose of the experiments which were made in the labo-

[1] D. W. Taylor, Experiments with Ventilating Fans and Pipes, Society of Naval Architects and Marine Engineers, November, 1905; Frank H. Kneeland, Trans. Am. Soc. M. E., vol. 33, p. 1137; G. F. Gebhardt, Journal Am. Soc. M. E., November, 1909; R. Burnham, Engineering News, December 21, 1905; Forrest M. Towl, Columbia University Lectures, 1911; Chas. H. Treat, Trans. Am. Soc. M. E., vol. 34, p. 1019; Thos. R. Weymouth, Trans. Am. Soc. M. E., vol. 34, p. 1094.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

ratories of the University of Wisconsin and which form the subject matter of this paper may be stated briefly as follows:

*a* To investigate the reliability of the pitot tube as a means of measuring gases.

*b* To ascertain which forms of the pitot tube now in common use give correct and which incorrect results in the measurement of gases.

4 It was planned to force air through a pipe in which the pitot tube to be tested was inserted, together with a standard gas meter,

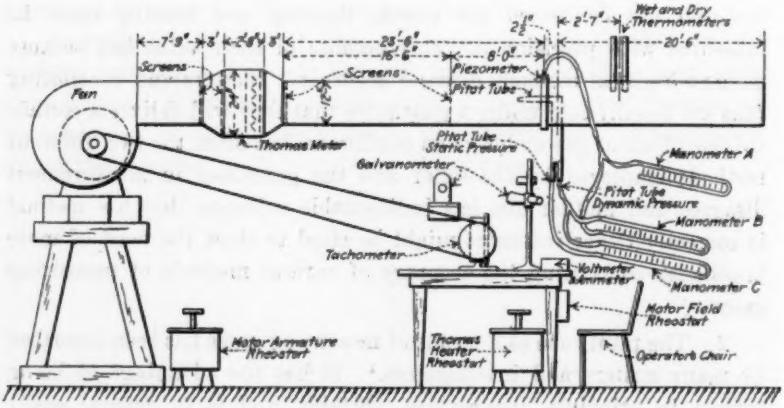



FIG. 1 SKETCH OF APPARATUS

known to be correct. This plan as a whole presented no difficulties except the selection of a standard gas meter concerning whose accuracy there could be no question.

5 The only fundamentally correct means of measuring directly the volume of a large quantity of flowing gas is the displacement or "holder" method. A holder of known dimensions drops a given distance, thus forcing a certain definite volume of gas through the discharge pipe, where it is again measured by the meter to be tested. The temperature, pressure and humidity of the gas both in the holder and at the apparatus to be tested must be known in order that both volumes may be reduced to the same conditions and thus a fair comparison may be made. It is a very difficult matter to obtain fair average readings of these quantities, especially of the temperature, because of the influence of the water in the holder, of the weather conditions outside, and of the large volume of gas in the holder.

There are certain periods in the spring and autumn when the temperature remains practically constant day and night for several days, and this is the only time when holder tests can be made with any approach to accuracy.

6 These considerations, as well as the fact that the holder is clumsy, intermittent in action and altogether unsuited for laboratory experiments, made its use as a standard of measurement out of the question.

7 The Thomas electric gas meter[1] which has been developed

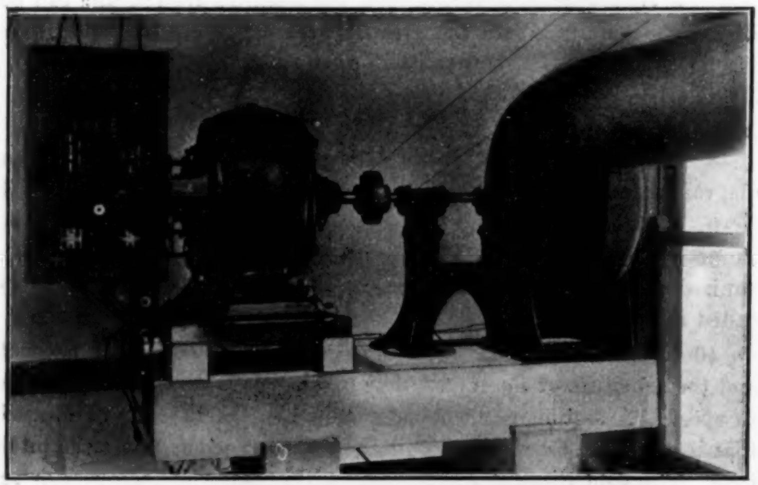


FIG. 2 VIEW OF FAN AND MOTOR

during the last few years and which is described in detail in a later portion of this paper, was used as the standard gas meter in the experiments with pitot tubes for the following reasons:

*a* It measures weight of gas, and avoids the difficulties inherent in volumetric measurements.

*b* Its measurement of gas depends directly upon the specific heat of a gas. The specific heat varies but slightly with wide changes in temperature, pressure and humidity.

*c* Its accuracy is limited only by the exactness of electrical

[1]Carl C. Thomas: Electric Gas Meter, The Journal, Am.Soc.M.E., December, 1909; Measurement of Gas, Journal of the Franklin Institute, November, 1911; Some Recent Developments in Gas Measuring Apparatus Proc., Am. Gas Inst., vol. 7, 1912.

measurement; such measurements can be made by engineers with a very great degree of refinement with well-known highly developed instruments.

*d* It has been thoroughly tested by experiments under the most varied conditions in this country and abroad; some of the results of tests are presented in Appendix No. 1, and in every instance it was proven that the meter was correct not only in theory but also in the actual measurement of gases.

*e* Its operation is very simple, the readings are few and can be obtained with the greatest accuracy, while it requires almost no attention itself when in use.

### DESCRIPTION OF THE APPARATUS

8 The apparatus used in the experiments on pitot tubes is shown diagrammatically in Fig. 1 and by photographs in Figs. 2, 3 and 4

9 A No. 2½ Sirocco fan driven by a 7½-h.p. direct-current shunt-wound motor forces air through the Thomas meter into a galvanized iron pipe 12 in. in diameter in which the pitot tube to be tested is inserted.

10 Variable resistances were placed in series with both the field and the armature of the motor, thus making possible a wide variation in speed. A stationary tachometer belted to the fan was so located that it could be observed at all times by the experimenter at the pitot tube. The field rheostat was brought within reach of the operator so that the fan could be maintained at any desired constant speed during a test.

11 Screens were inserted at the points shown in Fig. 1 in order to break up eddies and whirls and to have the air flow as nearly parallel as possible at the point where the pitot tube readings were taken. All joints between the Thomas meter and the pitot tube were made thoroughly air tight to prevent leakage. The barrel of the Thomas meter was lagged by three thicknesses of heavy blanket in order to prevent any possibility of error due to radiation to or from the meter casing.

12 The experimenter was stationed directly under the pitot tube and all readings were take at this point. A mercury barometer, hung on the adjacent wall, gave the atmospheric pressure, and a manometer inclined at a 10 to 1 slope made it possible to determine accurately

the static pressure in the pipe above atmosphere. The dry-bulb thermometer indicated the temperature of the air flowing in the pipe and together with the wet-bulb thermometer gave readings from which the humidity could be determined.

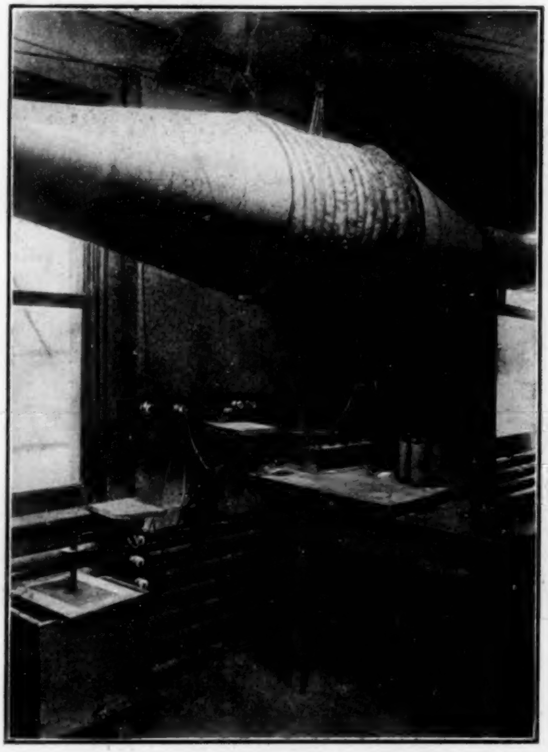


FIG. 3 VIEW OF THOMAS METER

THE THOMAS ELECTRIC METER

13 The Thomas electric meter is based on the principle of heating the gas through a known range of temperature and measuring the energy required to cause this change in temperature; this measured heat is proportional to the weight of gas flowing. Electric energy is used as the source of heat as it can be accurately measured and easily controlled. The temperature range is determined by the use of electrical resistance thermometers. If $E$ is the amount of energy in watts supplied to the heating coils to raise the temperature of $W$ lb.

of gas per minute through $t$ deg. fahr., and if $s$ is the specific heat at constant pressure of 1 lb. of gas, then

$$W = \frac{0.05686E}{ts}$$

14 The manually controlled Thomas meter used in these experiments is shown diagrammatically in Fig. 5. An electric heater

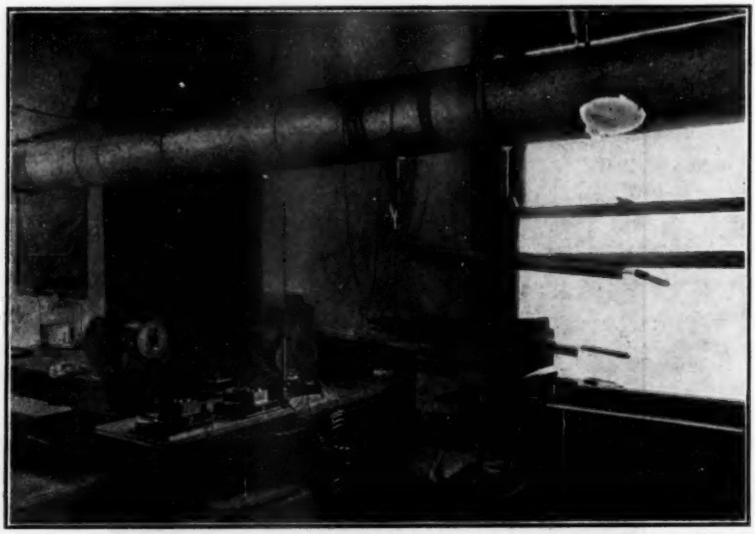


Fig. 4 View Taken at Observer's Station

is placed within a casing between two electric resistance thermometers $T_1$ and $T_2$. The heater consists of bare resistance wire mounted on a fiber frame and evenly distributed over the section of the casing. A water rheostat is placed in the heater circuit for regulating the direct-current supplied. The energy is measured by an ammeter and a voltmeter, both accurately calibrated in the standards laboratory of the electrical department of the University of Wisconsin after each series of tests.

15 The thermometers consist of nickel resistance wire wound on insulated wooden spindles which are evenly distributed over the cross-section of the casing. These thermometers were calibrated simultaneously by the author by means of a special apparatus at the plant of The Cutler-Hammer Manufacturing Com-

pany, Milwaukee, Wis., where the commercial form of the Thomas electric meter is manufactured. The results of this calibration are given by the curves in Fig. 26, Appendix No. 2. This curve shows the ohms increase in resistance per degree rise in temperature for any ordinary temperature. These two thermometers were arranged to form two arms of a Wheatstone bridge, of which the other two

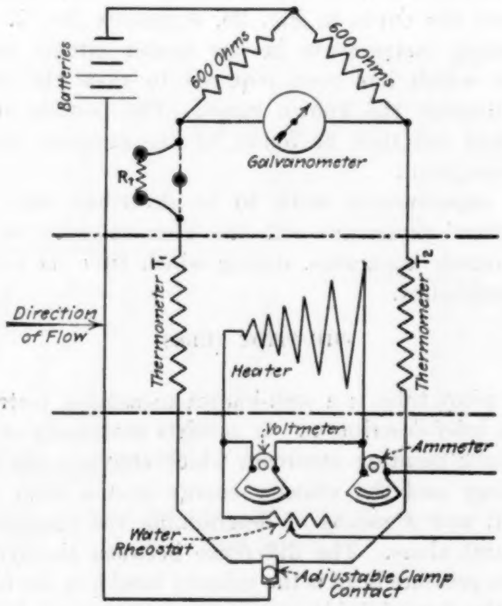



FIG. 5 DIAGRAM OF THOMAS ELECTRIC METER

arms were fixed resistance coils (600 ohms each), made of wire having a zero temperature coefficient. A galvanometer is connected across the Wheatstone bridge thus formed and an adjustable clamp contact is provided to balance the bridge when no heat is passing through the heater. A small resistance $R_t$, also with zero temperature coefficient, is arranged so that it can be placed in or out of series with the entrance thermometer. The resistance $R_t$ used in these experiments was 2.7998 ohms at all temperatures.

16 The operation of the meter is as follows: With gas flowing steadily through the meter but with no energy in the heater, and with $R_t$ out of circuit, the Wheatstone bridge is balanced by means of the adjustable clamp contact. Then the resistance $R_t$

is put in circuit and sufficient electrical energy is supplied to the heater to bring the galvanometer to a balance again. This balanced state can be attained only by bringing the exit gas to such a temperature above that of the entering gas that the resistance of thermometer $T_2$ has increased 2.7998 ohms above the resistance of the thermometer $T_1$. The required increase in temperature of the gas varies slightly with the average gas temperature and may be found from the curve in Fig. 26, Appendix No. 2. The electrical measuring instruments in the heater circuit indicate the energy input which has been required to raise the temperature of the gas through the known range. The pounds of gas flowing per minute can then be found by the equation given in the preceding description.

17 The experimental work to be described was done after about two years' experience with the Thomas meter as a piece of regular laboratory apparatus, during which time its accuracy had been fully established.

### THE PITOT TUBES

18 The pitot tube is a well-known measuring instrument and needs only a brief description. It consists essentially of two parts: a dynamic tube pointing upstream which converts the sum of the pressure energy and the velocity energy into a head which may be measured; and a means of determining the pressure head (or static pressure) alone. The difference between the dynamic head and the static pressure head is the velocity head $h$ in the fundamental formula for the flow of fluids

$$v = \sqrt{2\,gh}$$

where

$v$ = velocity in ft. per second
$g$ = 32.2 ft. per second per second
$h$ = mean velocity head in ft. of the fluid flowing

19 It has been satisfactorily proved and accepted that the dynamic tube gives the correct pressure if the tube points parallel to the current. But it is a very difficult matter to obtain the correct static pressure on account of secondary velocity effects. Therefore the study of the accuracy of the pitot tube resolves itself into a study of the correct method of obtaining the static pressure at the given cross-section where the tube is inserted.

20 Each pitot tube tested had as a part of the tube a means of determining the static pressure, and readings of the velocity head were obtained by using this pitot tube static pressure. Simultaneous readings of the velocity head were obtained by using a piezometer ring for the static pressure together with the dynamic tube of the pitot tube under test. This is further illustrated by reference to Fig. 1. Manometer *B* gives the velocity head by using the piezometer static

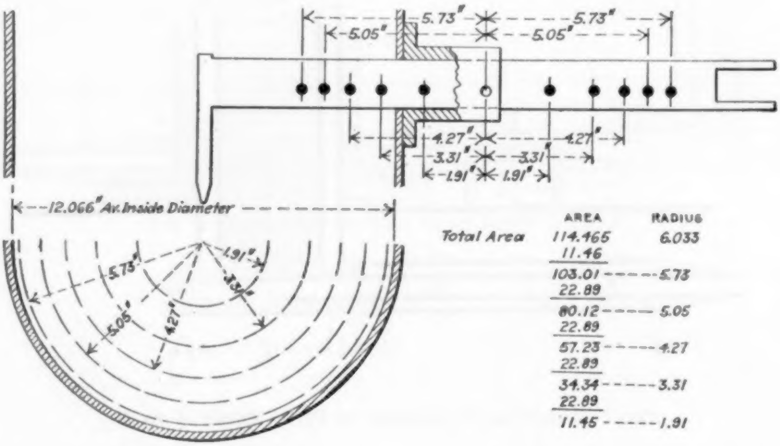



FIG. 6 SKETCH SHOWING POSITIONS AT WHICH READINGS OF VELOCITY HEAD WERE TAKEN

pressure, and the manometer *C* gives the velocity head by using the pitot tube static pressure. In both cases the same dynamic tube pressure is used. This piezometer, even if it were possible to be in error, would always indicate the same pressure under the same conditions irrespective of the tube under test and thus it afforded an additional means of *comparison* independent of the Thomas meter.

21 The piezometer is shown in Fig. 1 and Fig. 7 and is simply an absolutely air-tight annular space about the pipe, connected with the interior of the pipe by six small holes 0.04 in. in diameter.

22 The velocity of a gas flowing through a pipe is much greater at the center than near the walls of the pipe.[1] In addition, it was apparent from the tests that the gas flows through the pipe with a wave or spiral motion even when many screens are inserted to

[1]Loeb, Journal, American Society of Naval Engineers, 1912, p. 1115.

straighten out the stream lines. Therefore, it was necessary to take a large number of readings across two diameters of the pipe. The total area of the pipe was divided into five concentric annular areas

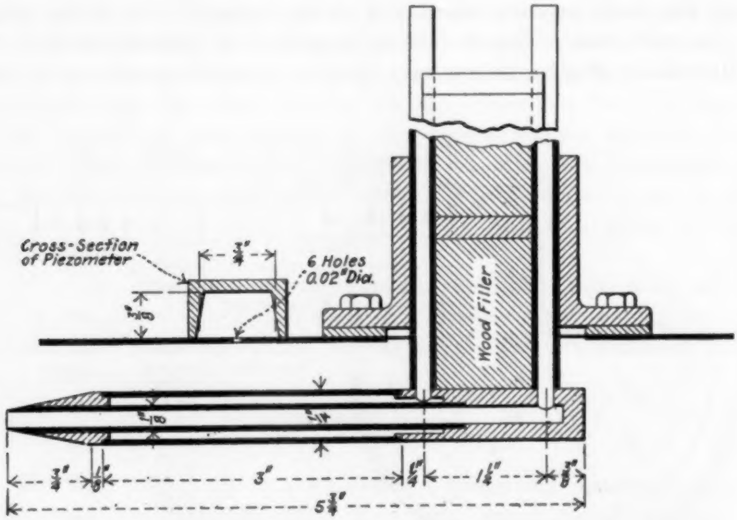



FIG. 7 DIMENSIONED SKETCH OF PITOT TUBES *A* TO *H*

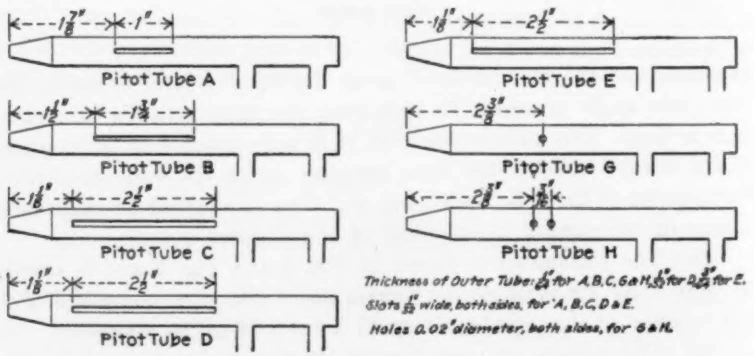



FIG. 8 SKETCHES OF PITOT TUBES *A* TO *H* SHOWING DIMENSIONS OF STATIC OPENINGS

and four readings of the velocity head were obtained in each test at the center of each annular area, thus giving 20 readings from which

the mean velocity head could be calculated. Since the velocity varies as the square root of the velocity head it was necessary to average the square roots of each of the 20 readings, and the square of this average represented the mean velocity head. The positions on the diameter at which the readings were taken are shown in Fig. 6. Readings were

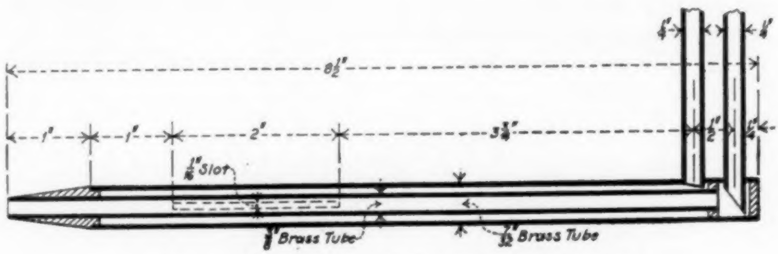


FIG. 9 DIMENSIONED SKETCH OF PITOT TUBE *X*

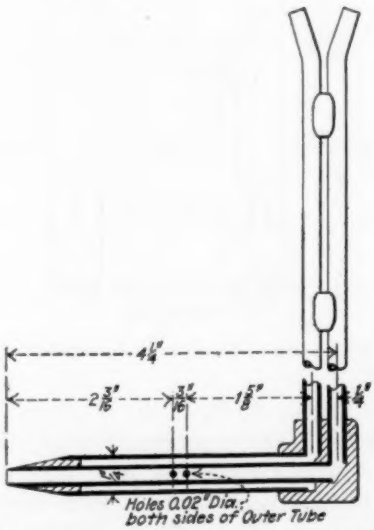


FIG. 10 DIMENSIONED SKETCH OF PITOT TUBE *Y*

also taken at the center of the pipe in order to determine if any definite relation existed between the mean velocity head and the velocity head at the center of the pipe.

23 The pitot tube under test was held in place by means of a brass bushing or holder which could be fastened by screws to brass

facings soldered to the outside of the galvanized air pipe. The facings were 90 deg. apart, thus permitting readings to be taken on both the horizontal and vertical diameters. Each pitot tube was first carefully centered in the pipe and a dowel hole bored in the shank corresponding to a hole in the holder. Ten other holes were bored in the shank at the proper distances either side of the center (see Fig. 6), so that by the use of a dowel pin the pitot tube could be quickly and accurately placed in its proper positions in the tube.

24 The pitot tubes tested in these experiments may be described

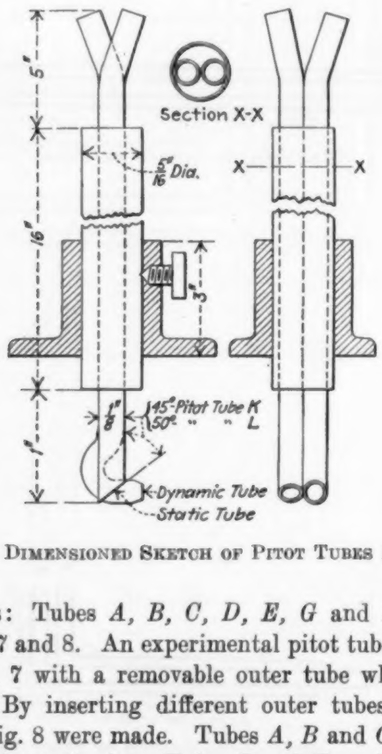



FIG. 11 DIMENSIONED SKETCH OF PITOT TUBES *K* AND *L*

briefly as follows: Tubes *A, B, C, D, E, G* and *H* are shown by sketches in Figs. 7 and 8. An experimental pitot tube was constructed as shown in Fig. 7 with a removable outer tube which contains the static opening. By inserting different outer tubes the seven pitot tubes shown in Fig. 8 were made. Tubes *A, B* and *C* are alike except for the length of the static slot. Tubes *D* and *E* are like tube *C* except for the thickness of the outer tube as shown in Fig. 8. Tubes *G* and *H* have small holes 0.02 in. in diameter for the static openings.

25 Tubes *X* and *Y* were loaned to the University of Wisconsin through the courtesy of Mr. F. R. Still of the American Blower Company. Tube *X,* Fig. 9, was made from a drawing furnished by

Captain D. W. Taylor,[1] U. S. N.; this form of tube is used as a standard for ventilation work by the United States Navy. Tube *Y*, Fig. 10, is the standard tube of the American Blower Company, and was developed by Mr. Chas. H. Treat, who has thoroughly tested the tube for accuracy.[2]

26 Tubes *K* and *L*, Fig. 11, were constructed from the author's drawings and were copied from descriptions of tubes used by Mr. Frank H. Kneeland.[2] These were alike except that the static opening

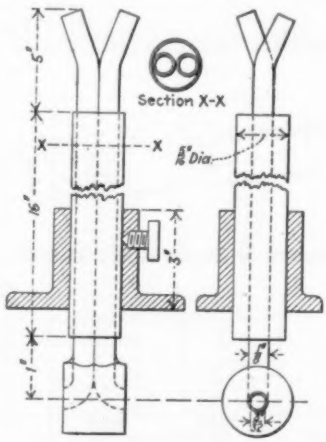



FIG. 12 DIMENSIONED SKETCH OF GERMAN STAUSCHEIBE

of the tube *K* was beveled to 45 deg., while in tube *L*, the static opening was flatter, being beveled to an angle of 50 deg. as shown in Fig. 11.

27 The "Stauscheibe,"[3] Fig. 12, has been widely used for several years in Germany and the experiments on this early form of pitot tube will be of interest for purposes of comparison.

### GAGES FOR MEASURING THE VELOCITY HEADS

28 The velocity heads measured were very small quantities, ranging from 0.08 in. to 1.6 in. of gasolene, so that gages of unusual

[1] Capt. D. W. Taylor; Soc. Naval Architects & Marine Engrs., Nov. 1905.

[2] Trans. Am. Soc. M. E., vol. 33, p. 1137.

[3] Reitschel; Versuche uber den Widerstand bei Bewegung der Luft in Rohrleitungen. Gesundheits Ingenieur, Festnummer July 1905. Marx; Uber die Messung von Luftgeschwindigkeiten. Gesundheits Ingenieur 1904.

accuracy had to be used. From previous experience it was known that the inclined manometer when properly constructed and carefully calibrated would give readings which were correct within the required limits of error. Two gages were therefore constructed as shown by the sketch in Fig. 13. The glass tubes were approximately 0.575 in. outside diameter and were selected with the greatest care from a large stock, special attention being given to straightness, uniformity of

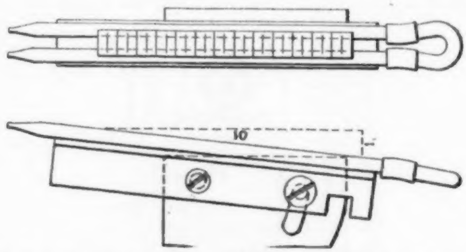



Fig. 13 Sketch of Inclined Manometers *A* and *B*

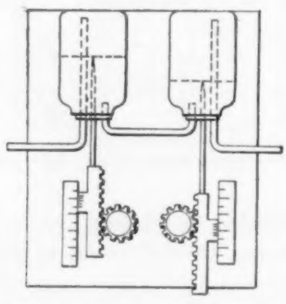


Fig. 14 Differential Hook Gage used as a Standard

bore and freedom from flaws. These gages were placed at a ten to one slope by means of an accurate template and a spirit level which was correct to 0.002 in. in 10 in. They were calibrated before and after each series of runs by means of the differential hook gage and micrometer, shown in Fig. 14. A few such representative calibrations are given in the Table 1.

29 A sliding scale made it possible to read the velocity head directly. Gasolene was used as a manometer fluid because it automatically kept the inside of the glass tubes clean, had a very definite meniscus and had almost no capillary attraction for the glass. Several

preliminary tests were run to determine whether the vapor tension of the gasolene vapor affected the readings. The manometers were first made to check each other when both contained gasolene and then the same pressure was measured when one manometer was filled with gasolene and the other with kerosene. Identical pressure readings were obtained with kerosene and gasolene. It, therefore, seemed

TABLE 1 SAMPLE CALIBRATIONS OF MANOMETERS *B* AND *C*

PRESSURES IN INCHES OF GASOLENE

| Differential Hook Gage | Manometer *C* | Manometer *B* | Differential Hook Gage | Manometer *C* | Manometer *B* |
|---|---|---|---|---|---|
| 1.872 | 1.863 | 1.857 | 1.530 | 1.531 | 1.513 |
| 1.657 | 1.658 | 1.654 | 1.382 | 1.387 | 1.375 |
| 1.463 | 1.461 | 1.451 | 1.216 | 1.214 | 1.215 |
| 1.285 | 1.278 | 1.274 | 1.051 | 1.053 | 1.045 |
| 1.071 | 1.076 | 1.071 | 0.948 | 0.956 | 0.949 |
| 0.877 | 0.878 | 0.877 | 0.836 | 0.837 | 0.830 |
| 0.768 | 0.773 | 0.770 | 0.727 | 0.735 | 0.730 |
| 0.665 | 0.669 | 0.667 | 0.503 | 0.508 | 0.502 |
| 0.560 | 0.570 | 0.568 | 0.397 | 0.400 | 0.397 |
| 0.462 | 0.468 | 0.460 | 0.286 | 0.290 | 0.284 |
| 0.368 | 0.369 | 0.367 | 0.224 | 0.225 | 0.220 |
| 0.175 | 0.170 | 0.168 | 0.103 | 0.109 | 0.105 |

Sp. Gr. Gasolene = 0.735 Sp. Gr. Gasolene = 0.736

evident that the vapor tension of either vapor in the manometer had no appreciable effect on the accuracy of the readings.

30 The large bore of the glass tubes (about ½ in. inside diameter) reduced any effect of capillary attraction to a negligible quantity and provided a reservoir of air which made the gages less sensitive to minor variations in the velocity head and therefore facilitated accurate reading.

### DESCRIPTIONS OF THE EXPERIMENTS

31 The general plan of the experiments was to calibrate each pitot tube against the Thomas meter under approximately similar conditions. Two series of eight or nine tests each were made with each tube, one series being made with the end of the pipe open (full gate), which provided the condition of high velocity with low static pressure; and the other series being made with the opening in the end of the pipe restricted to one-half the pipe area (full gate), thus pro-

viding the condition of low velocity and high static pressure. During each test the speed of the fan was kept constant for the length of time necessary to obtain all readings (20 to 30 minutes).

32 The procedure was as follows: The Thomas meter was "balanced" by causing air to flow through the pipe when there was no current flowing through the heater and the resistance *R*, (Fig. 5) was out of circuit; and by moving the adjustable clamp contact until the galvanometer came to a balance. Several half days were consumed in checking and rechecking the first balance point, but ordinarily 30 minutes preceding and following a series of tests was sufficient to show that it had not changed. At frequent intervals a half day's run was made to check the original balance point. After the Thomas meter was once balanced it was found that it was not necessary to move the adjustable clamp contact again, showing that the electrical apparatus was not affected appreciably by a change in temperature from 60 deg. to 100 deg. fahr.

33 Manometers *B* and *C*, Figs. 1 and 13, were filled with gasolene of known specific gravity and carefully adjusted until their readings agreed on the average with the differential hook gage, Fig. 14. The pitot tube was tested for leaks, placed in position in the pipe and properly connected to the manometers by small rubber tubing, after which this rubber tubing was tested for air leaks.

34 When all was in readiness the fan was brought to the speed desired for the first test, the resistance $R_t$ (Fig. 5) was placed in the thermometer circuit of the Thomas meter, the switch to the heater circuit was closed and the electrical energy to the heater regulated by the water rheostat until the galvanometer again came to a balance. Then keeping the speed constant and the galvanometer balanced, the pitot tube was placed successively at the proper points across the two diameters of the pipe and readings of the velocity head were obtained at each point. During the intervals when the manometers were coming to rest all necessary readings of pressure, temperature, revolutions per minute of fan, taken by a hand revolution counter, and volts and amperes in the heater circuit of the Thomas meter were obtained. The fact that the direct current used was furnished by a turbo-generator set having excellent voltage regulation and which supplied current for no other machines, contributed greatly to the constancy of conditions and to the ease of obtaining accurate data.

35 When all data had been obtained for the first test the speed

of the fan was increased until the tachometer needle pointed to the next determined speed, the current to the Thomas meter was regulated until the galvanometer balanced and then all readings were obtained as before.

36 At the end of a day's run the resistance $R_t$ (Fig. 5) was cut

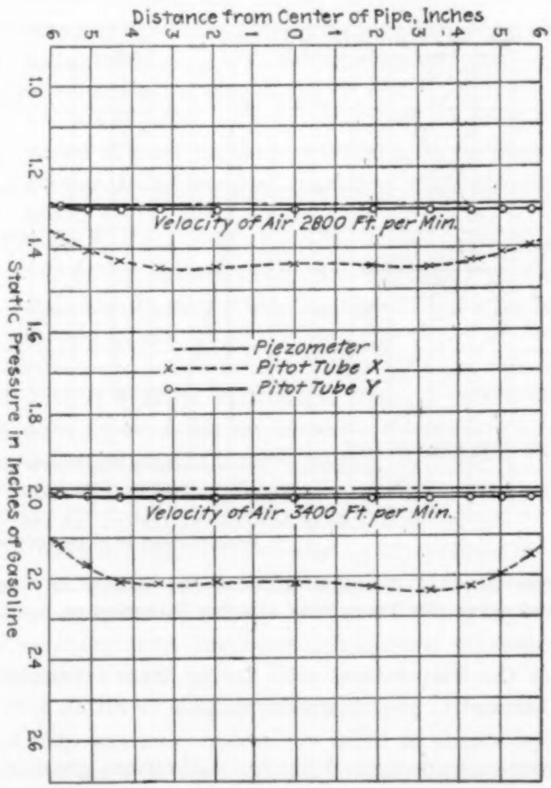



Fig. 15 Static Traverses of Pipe taken by Pitot Tubes $X$ and $Y$ under the Same Conditions of Velocity and Static Pressure

out of circuit and the Thomas meter balanced with no current flowing through the heater, the manometers $B$ and $C$ were calibrated by means of the differential hook gage and the gasolene emptied out of the manometers to check its specific gravity.

## LIMITS OF ACCURACY

37 The greatest possible error in the Thomas meter was the personal error of reading the volts and amperes to the heater and this error is estimated to be under 4/10 of 1 per cent. The thermometers used were carefully selected and calibrated by means of a standard thermometer from the Bureau of Standards, and were known to be accurate for the temperatures read during these experiments. The static and barometric pressures were readable to within 1/20 of 1 per cent.

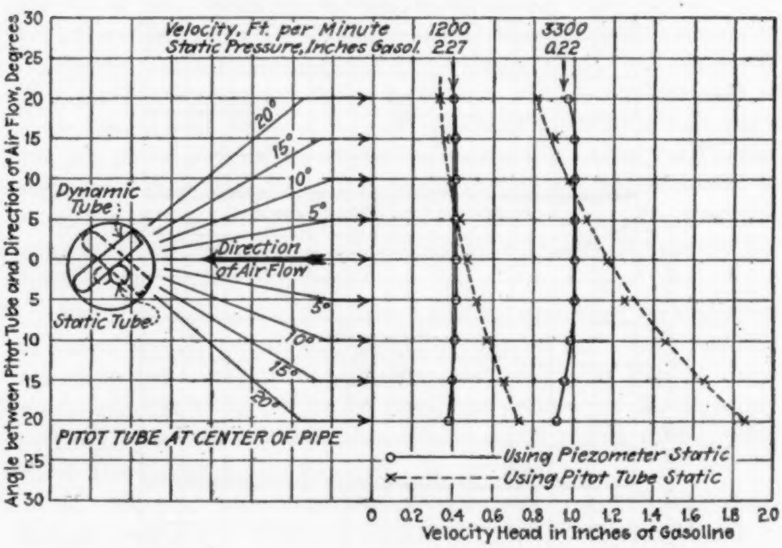



FIG. 16 PITOT TUBE *L*. DIAGRAM SHOWING THE EFFECT OF VARYING THE ANGLE BETWEEN THE PITOT TUBE AND THE DIRECTION OF AIR FLOW

38 From an inspection of the two calibrations given in Table 1, of the manometers *B* and *C* used to find the pitot tube velocity heads, it will be seen that the manometer readings may be in error on the average as much as 1 per cent. Since the velocity of air flowing through the pipe varies as the square root of the velocity head, an error of 1 per cent in the velocity head means an error of ½ of 1 per cent in the velocity itself.

39 The accuracy of the measuring instruments used is much greater than can be expected from the pitot tube as a means of

measuring gases, due to the uncontrollable variation in the air flow. The considerations which may prevent even an absolutely correct pitot tube from giving true results are as follows:

*a* The air flows through the pipe in a wave or spiral motion and at no time is the velocity uniformly distributed across the pipe, being greater in one quarter than in the other three quarters; the quarter of highest velocities may or may not be on the diameters where pitot tube readings are being taken.

*b* The velocities on the diameters where pitot tube readings are being taken may be constantly varying during the period of time necessary to obtain the readings; thus the average of all readings may be slightly too large or too small.

*c* The air flow can only approach, never reach, the ideal conditions of parallel flow, and the pitot tube is correct, theoretically, only when the tube is exactly parallel to the current of air.

40 From these considerations, together with the author's experience in this line of work, it is estimated that the results obtained in measuring gases by an absolutely correct pitot tube may vary 1 per cent, more or less, from the correct results. Of course, the average of a large number of tests should be more nearly correct than this, for the plus errors will probably neutralize the minus errors.

41 The Thomas meter is not affected by any of these irregularities in air flow because the resistance thermometers in the form of screens across the gas passage automatically integrate the varying temperature differences resulting from heating a non-uniform current of gas or air (see Appendix No. 1).

CALCULATIONS

42 Not only was every precaution taken to obtain correct data, but the calculations were made with the same degree of refinement. Since the obtaining of accurate results by both the Thomas meter and the pitot tube depended upon the use of correct values of the properties of air, a thorough study was made of these. As a result of this study the author devised and constructed the charts shown in Appendix No. 3, values taken from which agree almost exactly with the tables published by the Department of the Navy.[1] These charts are not only

[1] Dept. of Navy, Bureau of Construction and Repair, General Specifications, Appendix 8: Instructions for Calculating and Testing Ventilation System, 1908.

an exceedingly valuable aid in making calculations involving the properties of air, but also present these properties in what is thought to be a new graphical form. The basis of the calculations and the sources of information accompany the charts. The detailed calculations of the results of the experiments on pitot tubes are given in Appendix No. 2.

## RESULTS

43 The results of the experiments on pitot tubes are presented in Tables 2 to 15 inclusive and graphically in Figs. 17 to 25 inclusive. In order that the tabulated results may be thoroughly understood a brief explanation is necessary.

44 The test number, Column 26, is given for purposes of identification and is further explained in Appendix No. 2. The cubic feet of air per minute by all three methods, i.e., by the Thomas meter, by the pitot tube alone and by the pitot dynamic tube together with the piezometer, are given in columns 28, 29, 30, and are all reduced to the conditions of temperature, pressure and humidity as determined at the section where the pitot tube is inserted.

45 Columns 31 to 35 inclusive are explained by their use in the following formulae:

$$C_1 = MC_2, \text{ or } M = \frac{C_1}{C_2}$$

$$C_1 = NC_3, \text{ or } N = \frac{C_1}{C_3}$$

$$C_3 = QC_2, \text{ or } Q = \frac{C_3}{C_2}$$

$$V_1 = \sqrt{2gh_1} = \sqrt{2gUh_3}, \text{ or } U = \frac{h_1}{h_3}$$

$$V_2 = \sqrt{2gh_2} = \sqrt{2gZh_4}, \text{ or } Z = \frac{h_2}{h_4}$$

where

$C_1$ = cu. ft. of air per minute by Thomas meter

$C_2$ = cu. ft. of air per minute by the pitot tube using the pitot static pressure

$C_3$ = cu. ft. of air per minute by pitot dynamic tube and the piezometer static pressure

$V_1$ and $V_2$ = velocity of the air flowing in ft. per sec.

$h_1$ = mean velocity head in ft. of air flowing, obtained by the pitot tube using the pitot static pressure

$h_3$ = velocity head in ft. of air at the center of the pipe obtained in the same manner as $h_1$

$h_2$ = mean velocity head in ft. of air flowing, obtained by the pitot dynamic tube and the piezometer static pressure

$h_4$ = velocity head in ft. of air at the center of the pipe, obtained in the same manner as $h_2$

46 *M* and *N* are therefore coefficients by which the actual results obtained by the pitot tube must be multiplied to obtain the correct flow of gas. *Q* gives the coefficient by which the results from the pitot tube alone must be multiplied to obtain the same discharge as that shown by the pitot dynamic tube and the piezometer. *U* and *Z* state the relations between the velocity head at the center of the pipe and the mean velocity head as determined by the two methods used in the experiments.

47 From a study of the summary, Table 2, the following general results and conclusions may be stated:

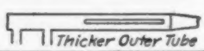
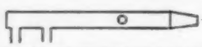
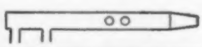
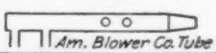
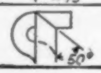
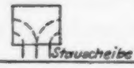
*a* The pitot tube as a means of measuring gases is reliable within approximately 1 per cent when the static pressure is correctly obtained and when all readings are taken with a sufficient degree of refinement; in order to obtain this degree of accuracy the pitot tube should be preceded by a length of pipe 20 to 38 times the pipe diameter in order to make the flow of gas as nearly uniform across the section of the pipe as possible.

*b* All the methods of obtaining the dynamic head used in these experiments, including the Stauscheibe, give accurate results.

*c* The most reliable and accurate means of obtaining the static pressure is the piezometer or its equivalent, the results of 138 separate tests using the piezometer static pressure agreeing with the Thomas meter within an average of 0.33 per cent; these results show beyond any doubt that the static pressure is constant across any section of a pipe in which gas is flowing at a uniform rate.

*d* Of the methods of obtaining the static pressure by the pitot tube itself, the most reliable and accurate is by means of a very small hole in a perfectly smooth surface, as in pitot tube *Y*.

TABLE 2 SUMMARY

| | Name of Tube | M | N | Q | U | Z |
|---|---|---|---|---|---|---|
|  | A | 1.0614 | 1.0175 | 1.0413 | 0.7696 | 0.7936 |
|  | B | ...... | ...... | 1.0576 | 0.8019 | 0.7898 |
|  | C | ...... | ...... | 1.0343 | 0.7996 | 0.7880 |
|  | *C | 1.0346 | 0.9952 | 1.0384 | 0.8107 | 0.8066 |
|   | D | ...... | ...... | 1.0369 | 0.7927 | 0.7942 |
|   | E | ...... | ...... | 1.0489 | 0.7885 | 0.7898 |
|   | X | 1.0987 | 0.9925 | 1.1074 | 0.7696 | 0.7867 |
|  | G | 1.0076 | 1.0085 | 0.9989 | 0.8100 | 0.8164 |
|  | H | 1.0218 | 1.0152 | 1.0065 | 0.7966 | 0.7957 |
|   | Y | 1.0024 | 1.0017 | 0.9992 | 0.8002 | 0.7996 |
|   | K | 1.0669 | 0.9924 | 1.0626 | 0.7617 | 0.8115 |
|   | L | 1.0104 | 1.0032 | 1.0064 | 0.7593 | 0.8112 |
| Average | | ...... | 1.0033 | ...... | 0.7884 | 0.7986 |
|   | S | 0.9861 | 1.0016 | 0.9844 | 0.7780 | 0.8228 |

*e* The long slots for obtaining the static pressure are not reliable and give results which are in error from 3.5 to 10 per cent. The fact that slots do not give correct results is further illustrated by Fig. 15. The length of the slots or the thickness of the outer tube does not appear to affect the accuracy of the tube.

*f* The beveled tube for obtaining the static pressure as used in pitot tubes $K$ and $L$ is not reliable. A very slight change in the angle of bevel produces an appreciable change in the result. In taking a traverse of a pipe the sides of the pipe affect the readings. But the greatest error is produced by the uncertainty as to whether the tube is pointing directly upstream. The effect of allowing the tube to point at an angle with the direction of flow is shown by Fig. 16, where it is seen that if the tube is off 20 deg. in one direction an error of 85 per cent in the velocity head is introduced.

*g* The Stauscheibe gives accurate results using either the static reading from the Stauscheibe and the special formula (Par. 79); or by using the piezometer static with the usual formula for the pitot tubes. In the first case the agreement is within 1.4 per cent and in the second within 0.16 per cent, as shown in Table 15 and Fig. 25.

*h* It appears that an approximate relation exists between the mean velocity head of a gas flowing through the pipe and the velocity head found by placing the tube at the center of the pipe. For a 12-in. galvanized iron pipe results within 2 per cent may be expected from using the formula

$$v = \sqrt{(2g)\ (0.80)\ h_c}$$

where

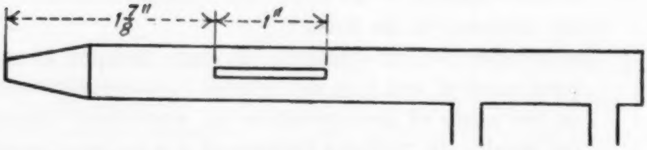
$v$ = velocity in ft. per second

$g$ = 32.2 ft. per second per second

$h_c$ = velocity head at center of pipe in ft. of gas flowing

48 The writer wishes to acknowledge the valuable help and suggestions given by Prof. Carl C. Thomas and Prof. A. G. Christie of the University of Wisconsin, and by Mr. J. C. Wilson, whose untiring interest made possible the success of these experiments. He also wishes to acknowledge his indebtedness to Mr. F. Lorig, who aided in obtaining many of the data, and to others who loaned apparatus or contributed suggestions.

TABLE 3 RESULTS OF PITOT TUBE *A*




| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | Col. 18 / Col. 20 | Col. 19 / Col. 21 |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *A*-1-F | 795 | 1710 | 1655 | 1680 | 1.043 | 1.017 | 1.015 | 0.773 | 0.790 |
| *A*-2-F | 870 | 1910 | 1828 | 1870 | 1.055 | 1.021 | 1.023 | 0.748 | 0.788 |
| *A*-3-F | 947 | 2085 | 1995 | 2058 | 1.045 | 1.013 | 1.030 | 0.767 | 0.789 |
| *A*-4-F | 1031 | 2265 | 2160 | 2237 | 1.047 | 1.013 | 1.034 | 0.769 | 0.784 |
| *A*-5-F | 1118 | 2487 | 2340 | 2440 | 1.063 | 1.019 | 1.043 | 0.768 | 0.792 |
| *A*-6-F | 1191 | 2675 | 2513 | 2620 | 1.064 | 1.020 | 1.043 | 0.775 | 0.793 |
| *A*-7-F | 1273 | 2885 | 2685 | 2805 | 1.074 | 1.028 | 1.044 | 0.773 | 0.788 |
| *A*-8-F | 1370 | 3095 | 2880 | 3005 | 1.074 | 1.029 | 1.044 | 0.780 | 0.787 |
| *A*-9-F | 1458 | 3300 | 3080 | 3240 | 1.070 | 1.017 | 1.051 | 0.788 | 0.798 |
| Average.. | ........ | ........ | ........ | ........ | 1.0595 | 1.0197 | 1.0363 | 0.7712 | 0.790 |
| *A*-1-½ | 794 | 1080 | 1036 | 1068 | 1.049 | 1.010 | 1.037 | 0.775 | 0.813 |
| *A*-2-½ | 875 | 1178 | 1138 | 1171 | 1.035 | 1.005 | 1.030 | 0.772 | 0.793 |
| *A*-3-½ | 951 | 1315 | 1240 | 1280 | 1.059 | 1.026 | 1.032 | 0.772 | 0.778 |
| *A*-4-½ | 1028 | 1420 | 1371 | 1399 | 1.035 | 1.014 | 1.020 | 0.793 | 0.792 |
| *A*-5-½ | 1107 | 1552 | 1452 | 1510 | 1.068 | 1.025 | 1.040 | 0.752 | 0.780 |
| *A*-6-½ | 1190 | 1673 | 1573 | 1662 | 1.063 | 1.005 | 1.056 | 0.775 | 0.810 |
| *A*-7-½ | 1266 | 1803 | 1663 | 1770 | 1.090 | 1.023 | 1.064 | 0.765 | 0.798 |
| *A*-8-½ | 1371 | 1942 | 1782 | 1902 | 1.090 | 1.020 | 1.066 | 0.768 | 0.805 |
| *A*-9-½ | 1447 | 2049 | 1892 | 2030 | 1.083 | 1.010 | 1.072 | 0.760 | 0.805 |
| Average.......... | | ........ | ........ | ........ | 1.0633 | 1.0153 | 1.0463 | 0.7680 | 0.7971 |
| Net Average...... | | ........ | ........ | ........ | 1.0614 | 1.0175 | 1.0413 | 0.7696 | 0.7938 |

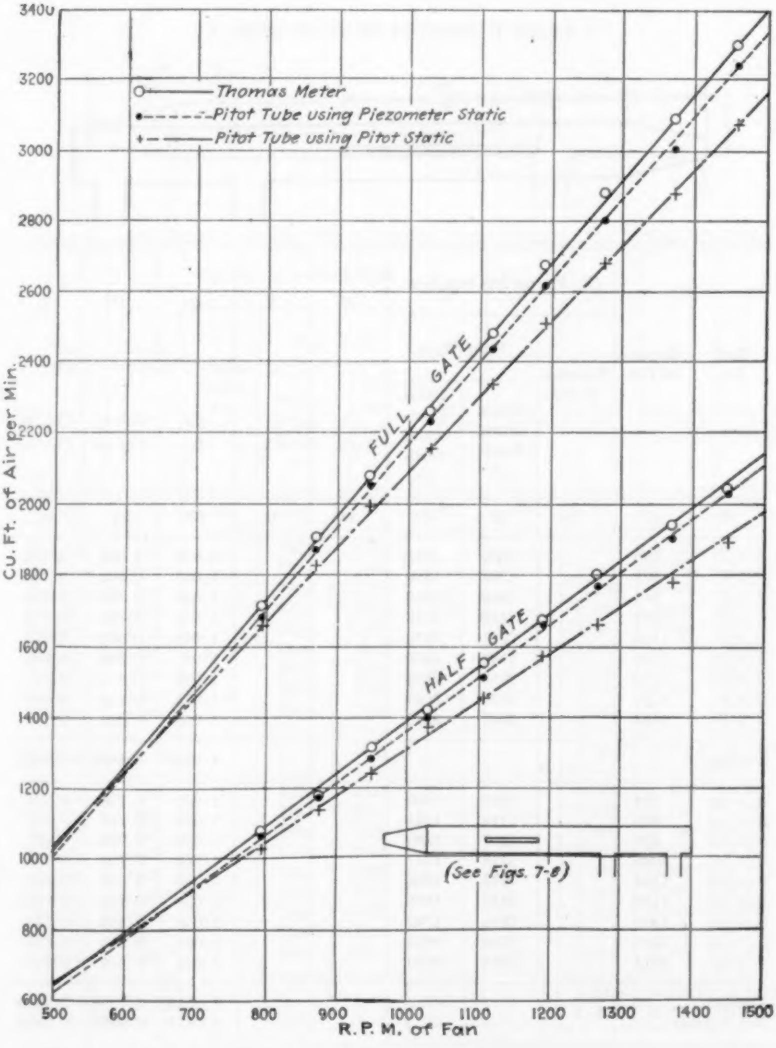
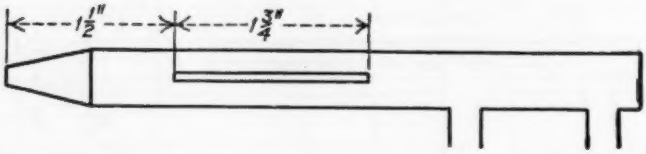


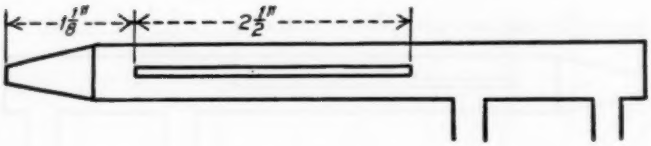

FIG. 17 PITOT TUBE *A*

TABLE 4 RESULTS OF PITOT TUBE *B*



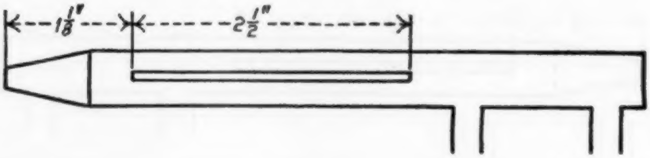

| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_3}$ | $\frac{C_1}{C_2}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *B*-1-F | 780 | .... | 1622 | 1714 | ..... | ..... | 1.056 | 0.802 | 0.792 |
| *B*-2-F | 877 | .... | 1792 | 1889 | ..... | ..... | 1.053 | 0.802 | 0.780 |
| *B*-3-F | 944 | .... | 1960 | 2080 | ..... | ..... | 1.061 | 0.798 | 0.788 |
| *B*-4-F | 1023 | .... | 2140 | 2265 | ..... | ..... | 1.058 | 0.797 | 0.779 |
| *B*-5-F | 1110 | .... | 2317 | 2470 | ..... | ..... | 1.058 | 0.800 | 0.790 |
| *B*-6-F | 1194 | .... | 2488 | 2660 | ..... | ..... | 1.068 | 0.808 | 0.792 |
| *B*-7-F | 1275 | .... | 2650 | 2835 | ..... | ..... | 1.069 | 0.812 | 0.787 |
| *B*-8-F | 1375 | .... | 2820 | 3040 | ..... | ..... | 1.076 | 0.813 | 0.793 |
| *B*-9-F | 1458 | .... | 3007 | 3240 | ..... | ..... | 1.076 | 0.815 | 0.795 |
| Average.. | ........ | .... | .... | .... | ..... | ..... | 1.0639 | 0.8052 | 0.7885 |
| *B*-1-½ | 784 | .... | 1020 | 1050 | ..... | ..... | 1.030 | 0.786 | 0.788 |
| *B*-2-½ | 870 | .... | 1118 | 1163 | ..... | ..... | 1.039 | 0.784 | 0.782 |
| *B*-3-½ | 950 | .... | 1217 | 1280 | ..... | ..... | 1.050 | 0.788 | 0.802 |
| *B*-4-½ | 1030 | .... | 1318 | 1377 | ..... | ..... | 1.045 | 0.792 | 0.778 |
| *B*-5-½ | 1104 | .... | 1432 | 1508 | ..... | ..... | 1.053 | 0.813 | 0.798 |
| *B*-6-½ | 1185 | .... | 1540 | 1628 | ..... | ..... | 1.057 | 0.810 | 0.790 |
| *B*-7-½ | 1266 | .... | 1648 | 1740 | ..... | ..... | 1.056 | 0.805 | 0.785 |
| *B*-8-½ | 1348 | .... | 1760 | 1875 | ..... | ..... | 1.064 | 0.810 | 0.798 |
| *B*-9-½ | 1447 | .... | 1883 | 2010 | ..... | ..... | 1.068 | 0.809 | 0.797 |
| Average.......... | | .... | .... | .... | ..... | ..... | 1.0513 | 0.7986 | 0.7910 |
| Net Average...... | | .... | .... | .... | ..... | ..... | 1.0576 | 0.8019 | 0.7898 |

TABLE 5 RESULTS OF PITOT TUBE *C*




| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_7$ | Using Piezometer Static $C_8$ | $\frac{C_1}{C_7}$ | $\frac{C_1}{C_8}$ | $\frac{C_8}{C_7}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *C*-1-F | 790 | .... | 1623 | 1672 | ..... | ..... | 1.030 | 0.793 | 0.777 |
| *C*-2-F | 872 | .... | 1793 | 1860 | ..... | ..... | 1.036 | 0.793 | 0.762 |
| *C*-3-F | 950 | .... | 1975 | 2042 | ..... | ..... | 1.034 | 0.807 | 0.795 |
| *C*-4-F | 1030 | .... | 2160 | 2228 | ..... | ..... | 1.032 | 0.800 | 0.782 |
| *C*-5-F | 1124 | .... | 2348 | 2432 | ..... | ..... | 1.034 | 0.804 | 0.789 |
| *C*-6-F | 1198 | .... | 2505 | 2600 | ..... | ..... | 1.038 | 0.799 | 0.786 |
| *C*-7-F | 1289 | .... | 2685 | 2780 | ..... | ..... | 1.034 | 0.803 | 0.789 |
| *C*-8-F | 1378 | .... | 2870 | 2987 | ..... | ..... | 1.041 | 0.806 | 0.793 |
| *C*-9-F | 1466 | .... | 3065 | 3195 | ..... | ..... | 1.042 | 0.804 | 0.793 |
| Average.. | ........ | .... | .... | .... | ..... | ..... | 1.0357 | 0.8010 | 0.7847 |
| *C*-1-½ | 793 | .... | 1024 | 1044 | ..... | ..... | 1.020 | 0.800 | 0.785 |
| *C*-2-½ | 866 | .... | 1131 | 1150 | ..... | ..... | 1.016 | 0.796 | 0.781 |
| *C*-3-½ | 947 | .... | 1239 | 1268 | ..... | ..... | 1.023 | 0.796 | 0.788 |
| *C*-4-½ | 1033 | .... | 1340 | 1380 | ..... | ..... | 1.031 | 0.807 | 0.793 |
| *C*-5-½ | 1115 | .... | 1450 | 1505 | ..... | ..... | 1.038 | 0.800 | 0.794 |
| *C*-6-½ | 1194 | .... | 1562 | 1620 | ..... | ..... | 1.038 | 0.797 | 0.789 |
| *C*-7-½ | 1271 | .... | 1672 | 1737 | ..... | ..... | 1.038 | 0.795 | 0.790 |
| *C*-8-½ | 1360 | .... | 1800 | 1872 | ..... | ..... | 1.041 | 0.798 | 0.795 |
| *C*-9-½ | 1450 | .... | 1920 | 2015 | ..... | ..... | 1.050 | 0.802 | 0.806 |
| Average........... | | .... | .... | .... | ..... | ..... | 1.0328 | 0.7983 | 0.7912 |
| Net Average...... | | .... | .... | .... | ..... | ..... | 1.0343 | 0.7996 | 0.7880 |

TABLE 6 RESULTS OF PITOT TUBE *C




| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | Col. 18 / Col. 20 | Col. 19 / Col. 21 |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *C-1-F | ... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *C-2-F | 850 | 1693 | 1657 | 1708 | 1.022 | 0.992 | 1.030 | 0.809 | 0.794 |
| *C-3-F | 915 | 1850 | 1818 | 1872 | 1.018 | 0.989 | 1.030 | 0.801 | 0.792 |
| *C-4-F | 994 | 2025 | 1970 | 2039 | 1.027 | 0.993 | 1.033 | 0.809 | 0.803 |
| *C-5-F | 1068 | 2202 | 2133 | 2210 | 1.032 | 0.996 | 1.034 | 0.808 | 0.800 |
| *C-6-F | 1150 | 2370 | 2297 | 2385 | 1.033 | 0.994 | 1.037 | 0.813 | 0.804 |
| *C-7-F | 1214 | 2540 | 2468 | 2560 | 1.029 | 0.994 | 1.036 | 0.817 | 0.808 |
| *C-8-F | 1290 | 2720 | 2633 | 2740 | 1.033 | 0.994 | 1.040 | 0.814 | 0.806 |
| *C-9-F | 1385 | 2935 | 2820 | 2955 | 1.040 | 0.994 | 1.027 | 0.815 | 0.812 |
| Average.. | ........ | .... | .... | .... | 1.0292 | 0.9932 | 1.0336 | 0.8107 | 0.8024 |
| *C-1-½ | ... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *C-2-½ | 847 | 1143 | 1105 | 1126 | 1.033 | 1.015 | 1.020 | 0.807 | 0.807 |
| *C-3-½ | 914 | 1233 | 1202 | 1239 | 1.025 | 0.996 | 1.030 | 0.808 | 0.805 |
| *C-4-½ | 987 | 1326 | 1290 | 1340 | 1.029 | 0.990 | 1.037 | 0.808 | 0.803 |
| *C-5-½ | 1066 | 1440 | 1386 | 1448 | 1.040 | 0.996 | 1.044 | 0.817 | 0.813 |
| *C-6-½ | 1145 | 1544 | 1479 | 1553 | 1.044 | 0.995 | 1.049 | 0.817 | 0.817 |
| *C-7-½ | 1214 | 1643 | 1575 | 1658 | 1.043 | 0.992 | 1.052 | 0.808 | 0.808 |
| *C-8-½ | 1298 | 1773 | 1675 | 1780 | 1.058 | 0.996 | 1.063 | 0.802 | 0.818 |
| *C-9-½ | 1386 | 1885 | 1800 | 1890 | 1.047 | 0.998 | 1.050 | 0.818 | 0.815 |
| Average............ | | .... | .... | .... | 1.0399 | 0.9972 | 1.0431 | 0.8106 | 0.8107 |
| Net Average...... | | .... | .... | .... | 1.0346 | 0.9952 | 1.0384 | 0.8107 | 0.8066 |

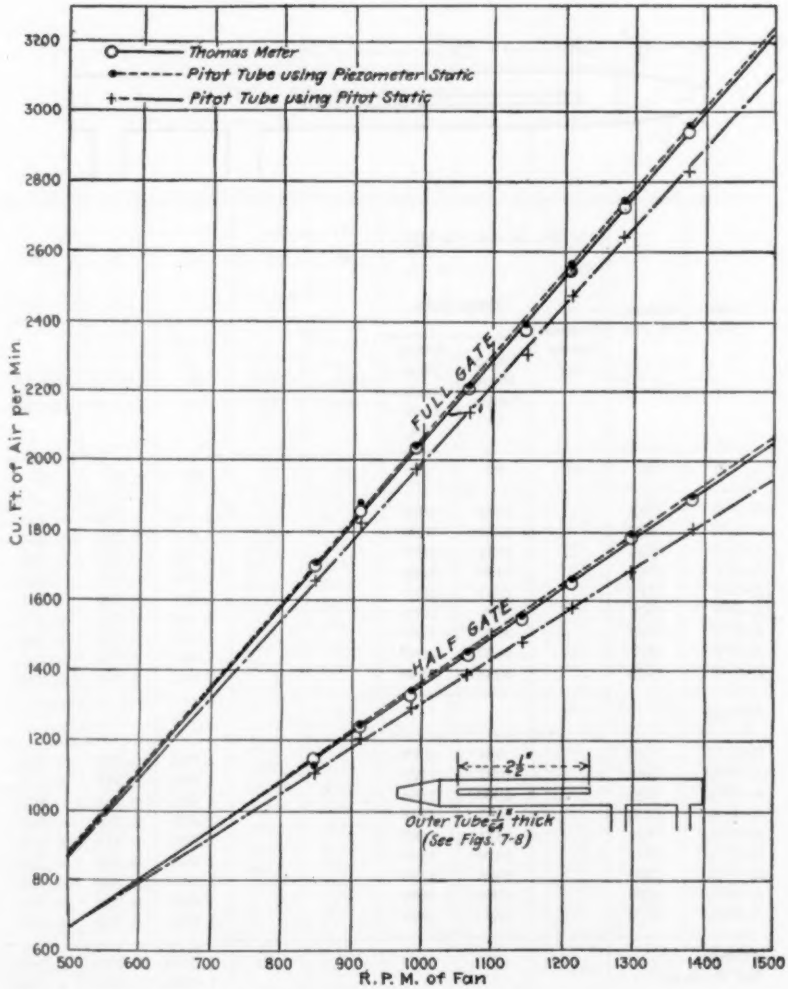
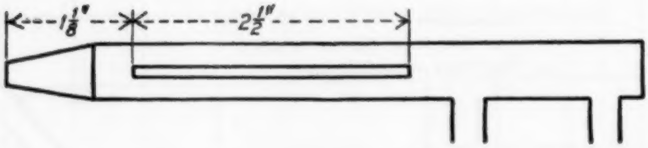


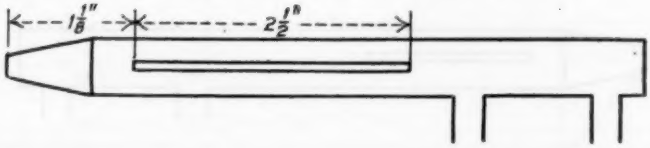

Fig. 18 Pitot Tube* *C*

TABLE 7 RESULTS OF PITOT TUBE *D*



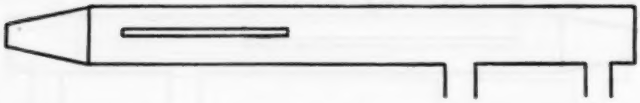

| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. — Thomas Meter $C_1$ | Cu. Ft. of Air per Min. — Pitot Tube — Using Pitot Static $C_2$ | Cu. Ft. of Air per Min. — Pitot Tube — Using Piezometer Static $C_3$ | M $\frac{C_1}{C_2}$ | N $\frac{C_1}{C_3}$ | Q $\frac{C_3}{C_2}$ | U $\frac{\text{Col. 18}}{\text{Col. 20}}$ | Z $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
|---|---|---|---|---|---|---|---|---|---|
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *D*-1-F | 792 | .... | 1583 | 1620 | ..... | ..... | 1.024 | 0.779 | 0.754 |
| *D*-2-F | 873 | .... | 1752 | 1805 | ..... | ..... | 1.030 | 0.789 | 0.777 |
| *D*-3-F | 954 | .... | 1932 | 2005 | ..... | ..... | 1.037 | 0.790 | 0.792 |
| *D*-4-F | 1029 | .... | 2100 | 2160 | ..... | ..... | 1.028 | 0.793 | 0.781 |
| *D*-5-F | 1118 | .... | 2295 | 2367 | ..... | ..... | 1.031 | 0.792 | 0.781 |
| *D*-6-F | 1202 | .... | 2470 | 2550 | ..... | ..... | 1.032 | 0.795 | 0.788 |
| *D*-7-F | 1283 | .... | 2660 | 2750 | ..... | ..... | 1.033 | 0.796 | 0.790 |
| *D*-8-F | 1377 | .... | 2850 | 2950 | ..... | ..... | 1.034 | 0.798 | 0.792 |
| *D*-9-F | 1465 | .... | 3045 | 3150 | ..... | ..... | 1.035 | 0.798 | 0.798 |
| Average........ | | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *D*-1-½ | 792 | .... | 984 | 1019 | ..... | ..... | 1.035 | 0.772 | 0.782 |
| *D*-2-½ | 871 | .... | 1097 | 1127 | ..... | ..... | 1.034 | 0.787 | 0.790 |
| *D*-3-½ | 955 | .... | 1200 | 1254 | ..... | ..... | 1.045 | 0.795 | 0.808 |
| *D*-4-½ | 1030 | .... | 1303 | 1348 | ..... | ..... | 1.035 | 0.790 | 0.786 |
| *D*-5-½ | 1121 | .... | 1431 | 1487 | ..... | ..... | 1.039 | 0.812 | 0.807 |
| *D*-6-½ | 1204 | .... | 1526 | 1593 | ..... | ..... | 1.043 | 0.795 | 0.792 |
| *D*-7-½ | 1290 | .... | 1635 | 1710 | ..... | ..... | 1.045 | 0.798 | 0.796 |
| *D*-8-½ | 1379 | .... | 1760 | 1850 | ..... | ..... | 1.050 | 0.793 | 0.795 |
| *D*-9-½ | 1473 | .... | 1880 | 1985 | ..... | ..... | 1.054 | 0.796 | 0.796 |
| Average........... | | .... | .... | .... | ..... | ..... | 1.0422 | 0.7931 | 0.7947 |
| Net Average....... | | .... | .... | .... | ..... | ..... | 1.0369 | 0.7927 | 0.7942 |

TABLE 8 RESULTS OF PITOT TUBE *E*




| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *E*-1-F | 790 | .... | 1577 | 1626 | ..... | ..... | 1.031 | 0.774 | 0.779 |
| *E*-2-F | 872 | .... | 1730 | 1797 | ..... | ..... | 1.039 | 0.779 | 0.781 |
| *E*-3-F | 951 | .... | 1895 | 1975 | ..... | ..... | 1.042 | 0.783 | 0.786 |
| *E*-4-F | 1031 | .... | 2060 | 2153 | ..... | ..... | 1.044 | 0.790 | 0.791 |
| *E*-5-F | 1116 | .... | 2242 | 2355 | ..... | ..... | 1.050 | 0.787 | 0.787 |
| *E*-6-F | 1202 | .... | 2418 | 2543 | ..... | ..... | 1.053 | 0.788 | 0.788 |
| *E*-7-F | 1282 | .... | 2592 | 2735 | ..... | ..... | 1.054 | 0.788 | 0.790 |
| *E*-8-F | 1380 | .... | 2850 | 2960 | ..... | ..... | 1.039 | 0.792 | 0.793 |
| *E*-9-F | 1465 | .... | 3050 | 3155 | ..... | ..... | 1.033 | 0.791 | 0.795 |
| Average | | .... | .... | .... | ..... | ..... | 1.0429 | 0.7859 | 0.7879 |
| *E*-1-½ | 792 | .... | 1008 | 1057 | ..... | ..... | 1.050 | 0.790 | 0.805 |
| *E*-2-½ | 873 | .... | 1130 | 1176 | ..... | ..... | 1.040 | 0.788 | 0.802 |
| *E*-3-½ | 952 | .... | 1224 | 1276 | ..... | ..... | 1.042 | 0.785 | 0.799 |
| *E*-4-½ | 1030 | .... | 1324 | 1385 | ..... | ..... | 1.046 | 0.785 | 0.795 |
| *E*-5-½ | 1117 | .... | 1440 | 1515 | ..... | ..... | 1.052 | 0.788 | 0.803 |
| *E*-6-½ | 1201 | .... | 1543 | 1630 | ..... | ..... | 1.056 | 0.790 | 0.798 |
| *E*-7-½ | 1284 | .... | 1655 | 1760 | ..... | ..... | 1.063 | 0.790 | 0.797 |
| *E*-8-½ | 1378 | .... | 1777 | 1895 | ..... | ..... | 1.065 | 0.800 | 0.808 |
| *E*-9-½ | 1468 | .... | 1890 | 2040 | ..... | ..... | 1.079 | 0.803 | 0.818 |
| Average | | .... | .... | .... | ..... | ..... | 1.0549 | 0.7910 | 0.7917 |
| Net Average | | .... | .... | .... | ..... | ..... | 1.0489 | 0.7885 | 0.7898 |

TABLE 9 RESULTS OF PITOT TUBE *X*



| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | *M* | *N* | *Q* | *U* | *Z* |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *X*-1-F | 807 | 1605 | 1510 | 1650 | 1.062 | 0.973 | 1.092 | 0.798 | 0.790 |
| *X*-2-F | 888 | 1800 | 1655 | 1825 | 1.087 | 0.987 | 1.102 | 0.794 | 0.793 |
| *X*-3-F | 963 | 1962 | 1805 | 2000 | 1.087 | 0.982 | 1.108 | 0.793 | 0.795 |
| *X*-4-F | 1046 | 2140 | 1975 | 2180 | 1.083 | 0.983 | 1.103 | 0.793 | 0.788 |
| *X*-5-F | 1135 | 2345 | 2125 | 2380 | 1.102 | 0.985 | 1.120 | 0.770 | 0.793 |
| *X*-6-F | 1229 | 2545 | 2280 | 2562 | 1.115 | 0.993 | 1.123 | 0.775 | 0.792 |
| *X*-7-F | 1311 | 2745 | 2455 | 2740 | 1.117 | 1.002 | 1.116 | 0.772 | 0.779 |
| *X*-8-F | 1410 | 2940 | 2625 | 2970 | 1.120 | 0.991 | 1.130 | 0.774 | 0.800 |
| *X*-9-F | 1496 | 3145 | 2785 | 3165 | 1.128 | 0.993 | 1.135 | 0.777 | 0.798 |
| Average........ | | .... | .... | .... | 1.1001 | 0.9877 | 1.1143 | 0.7830 | 0.7920 |
| *X*-1-½ | 785 | 1047 | 958 | 1037 | 1.093 | 1.010 | 1.083 | 0.738 | 0.767 |
| *X*-2-½ | 876 | 1161 | 1060 | 1153 | 1.095 | 1.007 | 1.088 | 0.751 | 0.788 |
| *X*-3-½ | 954 | 1270 | 1156 | 1260 | 1.098 | 1.007 | 1.090 | 0.757 | 0.790 |
| *X*-4-½ | 1028 | 1378 | 1256 | 1380 | 1.098 | 0.999 | 1.098 | 0.755 | 0.780 |
| *X*-5-½ | 1120 | 1490 | 1363 | 1488 | 1.094 | 1.001 | 1.093 | 0.755 | 0.770 |
| *X*-6-½ | 1195 | 1595 | 1457 | 1609 | 1.095 | 0.992 | 1.103 | 0.758 | 0.790 |
| *X*-7-½ | 1288 | 1713 | 1557 | 1723 | 1.100 | 0.994 | 1.105 | 0.760 | 0.782 |
| *X*-8-½ | 1370 | 1834 | 1670 | 1870 | 1.099 | 0.983 | 1.120 | 0.765 | 0.772 |
| *X*-9-½ | 1460 | 1960 | 1775 | 1996 | 1.103 | 0.983 | 1.124 | 0.767 | 0.793 |
| Average........... | | .... | .... | .... | 1.0972 | 0.9973 | 1.1004 | 0.7562 | 0.7813 |
| Net Average....... | | .... | .... | .... | 1.0987 | 0.9925 | 1.1074 | 0.7696 | 0.7867 |

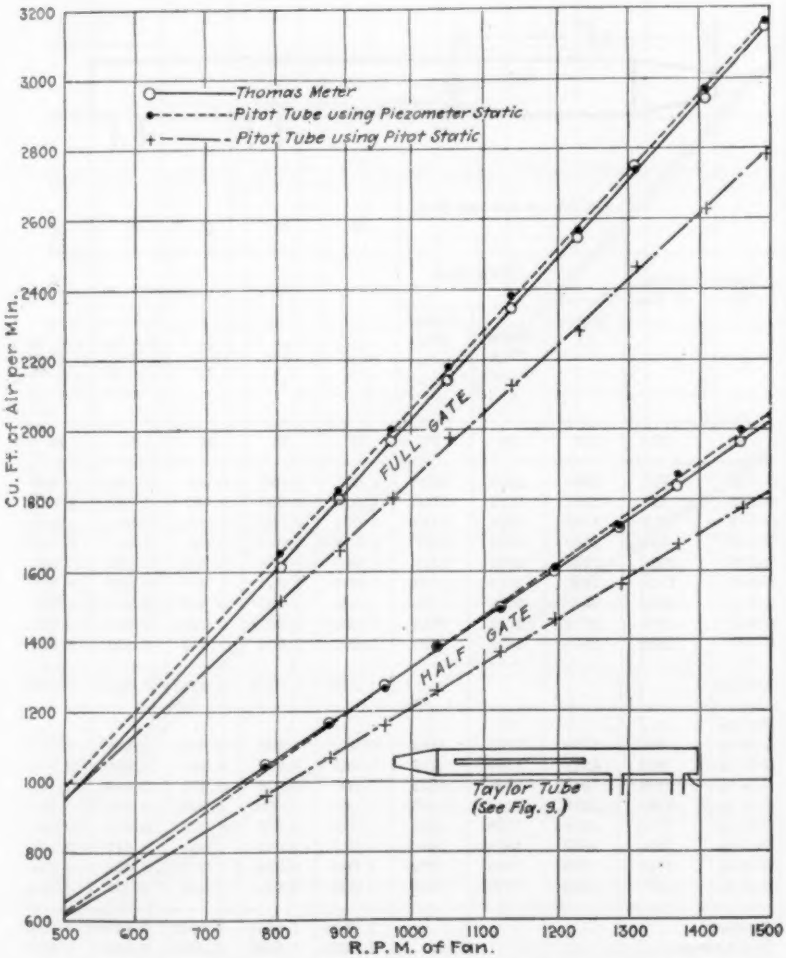
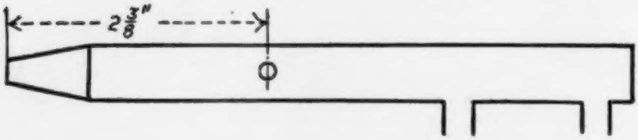



FIG. 19 PITOT TUBE *X*

TABLE 10 RESULTS OF PITOT TUBE *G*




| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *G*-1-F | 760 | 1590 | 1556 | 1550 | 1.021 | 1.026 | 0.997 | 0.798 | 0.806 |
| *G*-2-F | 835 | 1764 | 1722 | 1716 | 1.023 | 1.027 | 0.997 | 0.792 | 0.797 |
| *G*-3-F | 912 | 1912 | 1882 | 1870 | 1.021 | 1.022 | 0.994 | 0.811 | 0.812 |
| *G*-4-F | 984 | 2060 | 2032 | 2027 | 1.013 | 1.014 | 0.998 | 0.803 | 0.793 |
| *G*-5-F | 1060 | 2220 | 2220 | 2215 | 1.000 | 1.002 | 0.998 | 0.802 | 0.797 |
| *G*-6-F | 1128 | 2406 | 2370 | 2370 | 1.014 | 1.014 | 1.000 | 0.802 | 0.812 |
| *G*-7-F | 1202 | 2543 | 2537 | 2535 | 1.003 | 1.003 | 0.999 | 0.811 | 0.817 |
| *G*-8-F | 1278 | 2718 | 2718 | 2715 | 1.000 | 1.000 | 0.999 | 0.813 | 0.822 |
| *G*-9-F | 1356 | 2910 | 2895 | 2900 | 1.004 | 1.003 | 1.002 | 0.812 | 0.822 |
| Average | ........ | .... | .... | .... | 1.0110 | 1.0123 | 0.9982 | 0.8049 | 0.8097 |
| *G*-1-½ | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *G*-2-½ | 836 | 1124 | 1135 | 1114 | 0.992 | 1.009 | 0.982 | 0.811 | 0.812 |
| *G*-3-½ | 900 | 1217 | 1240 | 1227 | 0.982 | 0.993 | 0.991 | 0.805 | 0.814 |
| *G*-4-½ | 988 | 1343 | 1331 | 1323 | 1.007 | 1.013 | 0.993 | 0.812 | 0.818 |
| *G*-5-½ | 1060 | 1439 | 1435 | 1430 | 1.002 | 1.006 | 0.998 | 0.809 | 0.813 |
| *G*-6-½ | 1135 | 1554 | 1538 | 1544 | 1.010 | 1.006 | 1.003 | 0.813 | 0.823 |
| *G*-7-½ | 1210 | 1663 | 1632 | 1640 | 1.017 | 1.013 | 1.004 | 0.811 | 0.821 |
| *G*-8-½ | 1288 | 1769 | 1753 | 1772 | 1.009 | 0.998 | 1.010 | 0.822 | 0.834 |
| *G*-9-½ | 1370 | 1883 | 1870 | 1885 | 1.005 | 0.999 | 1.006 | 0.837 | 0.849 |
| Average | ........ | .... | .... | .... | 1.0042 | 1.0046 | 0.9995 | 0.8150 | 0.8230 |
| Net Average | ...... | .... | .... | .... | 1.0076 | 1.0085 | 0.9989 | 0.8100 | 0.8164 |

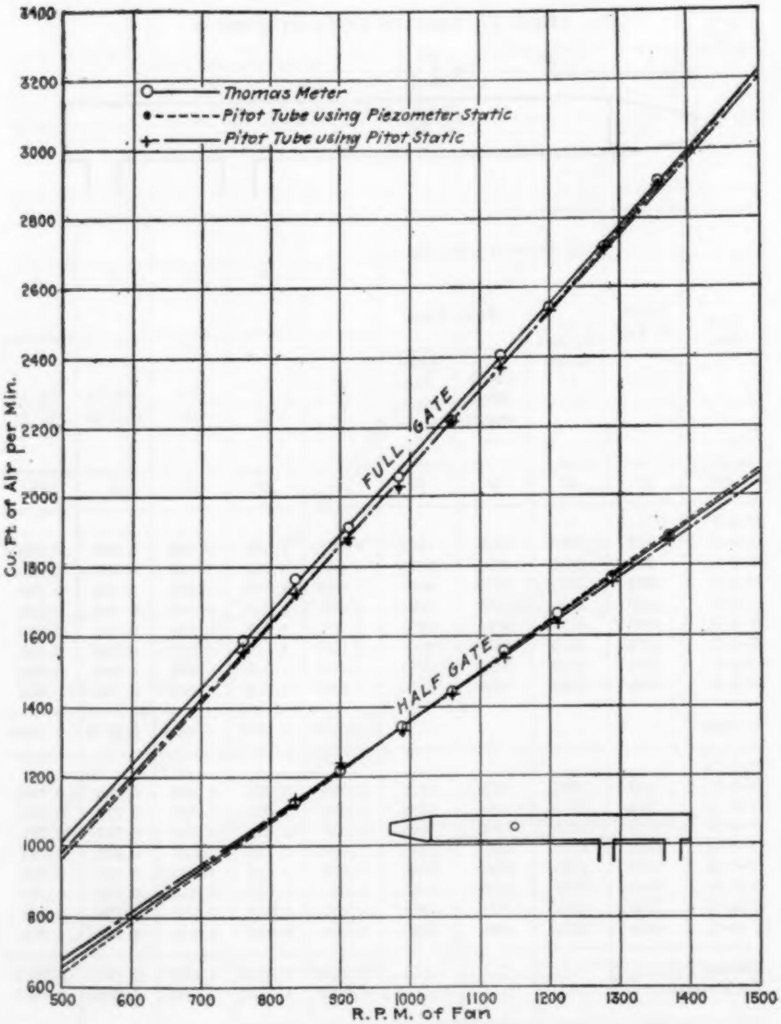
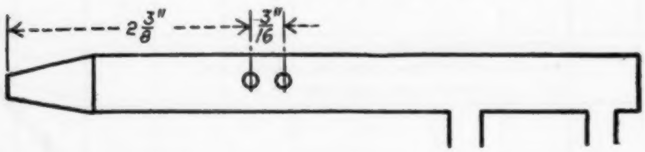



FIG. 20 PITOT TUBE *G*

TABLE 11 RESULTS OF PITOT TUBE *H*




| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | *M* | *N* | *Q* | *U* | *Z* |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | Col. 18 / Col. 20 | Col. 19 / Col. 21 |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *H*-1-F | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *H*-2-F | 872 | 1930 | 1922 | 1912 | 1.003 | 1.008 | 0.998 | 0.803 | 0.798 |
| *H*-3-F | 947 | 2108 | 2123 | 2123 | 0.993 | 0.993 | 1.000 | 0.830 | 0.826 |
| *H*-4-F | 1024 | 2300 | 2275 | 2269 | 1.010 | 1.013 | 0.998 | 0.796 | 0.790 |
| *H*-5-F | 1110 | 2508 | 2465 | 2463 | 1.017 | 1.018 | 0.999 | 0.795 | 0.792 |
| *H*-6-F | 1195 | 2710 | 2656 | 2653 | 1.016 | 1.017 | 0.999 | 0.792 | 0.790 |
| *H*-7-F | 1273 | 2910 | 2855 | 2855 | 1.019 | 1.019 | 1.000 | 0.799 | 0.798 |
| *H*-8-F | 1375 | 3120 | 3062 | 3060 | 1.018 | 1.019 | 0.999 | 0.800 | 0.800 |
| *H*-9-F | 1456 | 3330 | 3265 | 3270 | 1.019 | 1.018 | 1.001 | 0.798 | 0.803 |
| Average........ | | .... | .... | .... | 1.0120 | 1.0131 | 0.9992 | 0.8016 | 0.7996 |
| *H*-1-½ | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *H*-2-½ | 870 | 1203 | 1170 | 1176 | 1.026 | 1.023 | 1.006 | 0.788 | 0.793 |
| *H*-3-½ | 948 | 1313 | 1274 | 1284 | 1.028 | 1.021 | 1.007 | 0.775 | 0.780 |
| *H*-4-½ | 1029 | 1425 | 1390 | 1403 | 1.025 | 1.015 | 1.010 | 0.787 | 0.787 |
| *H*-5-½ | 1110 | 1570 | 1523 | 1535 | 1.030 | 1.022 | 1.007 | 0.802 | 0.791 |
| *H*-6-½ | 1185 | 1685 | 1623 | 1654 | 1.036 | 1.018 | 1.020 | 0.797 | 0.795 |
| *H*-7-½ | 1265 | 1793 | 1738 | 1763 | 1.032 | 1.016 | 1.013 | 0.796 | 0.791 |
| *H*-8-½ | 1360 | 1932 | 1870 | 1910 | 1.033 | 1.010 | 1.021 | 0.798 | 0.801 |
| *H*-9-½ | 1454 | 2072 | 1986 | 2040 | 1.043 | 1.013 | 1.026 | 0.789 | 0.796 |
| Average.......... | | .... | .... | .... | 1.0316 | 1.0172 | 1.0137 | 0.7915 | 0.7917 |
| Net Average...... | | .... | .... | .... | 1.0218 | 1.0152 | 1.0065 | 0.7966 | 0.7957 |

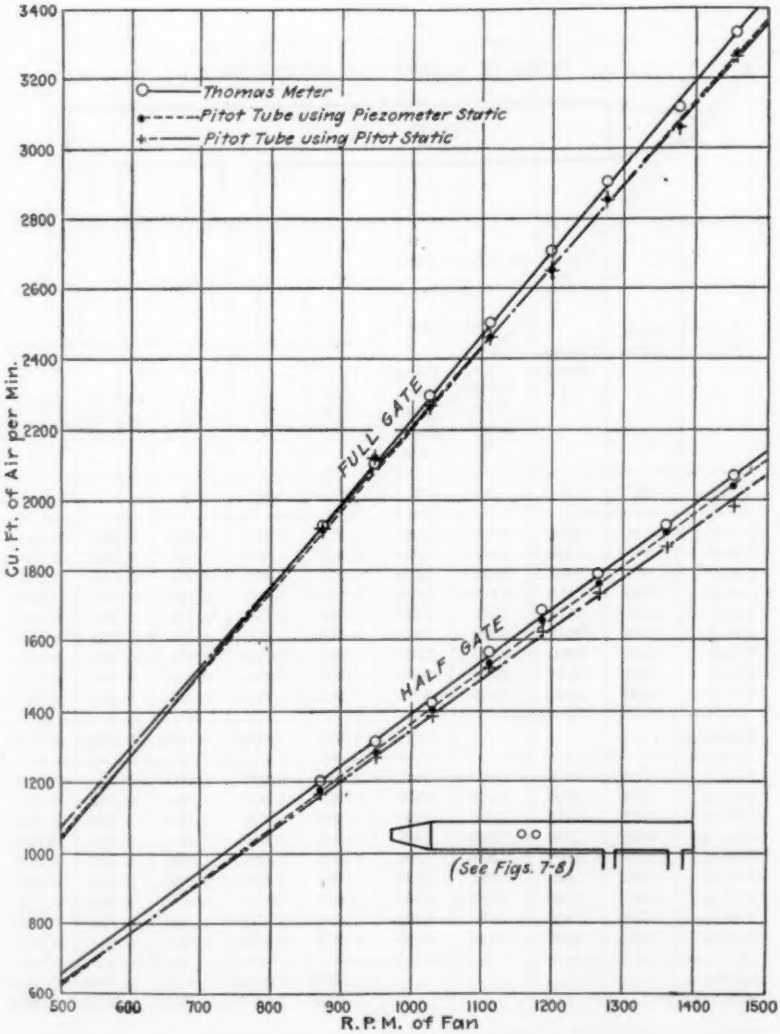
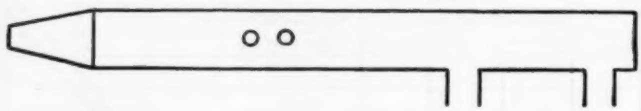



Fig. 21 Pitot Tube *H*

TABLE 12 RESULTS OF PITOT TUBE *Y*



| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | *M* | *N* | *Q* | *U* | *Z* |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | Col. 18 / Col. 20 | Col. 19 / Col. 21 |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *Y*–1–F | 793 | 1635 | 1652 | 1650 | 0.991 | 0.992 | 0.999 | 0.800 | 0.813 |
| *Y*–2–F | 887 | 1865 | 1860 | 1860 | 1.002 | 1.002 | 1.000 | 0.799 | 0.808 |
| *Y*–3–F | 968 | 2033 | 2032 | 2030 | 1.000 | 1.001 | 0.999 | 0.803 | 0.796 |
| *Y*–4–F | 1050 | 2215 | 2215 | 2205 | 1.000 | 1.003 | 0.997 | 0.797 | 0.812 |
| *Y*–5–F | 1144 | 2424 | 2425 | 2422 | 1.000 | 1.001 | 0.999 | 0.801 | 0.795 |
| *Y*–6–F | 1226 | 2606 | 2608 | 2595 | 0.999 | 1.003 | 0.996 | 0.805 | 0.797 |
| *Y*–7–F | 1316 | 2800 | 2800 | 2790 | 1.000 | 1.003 | 0.998 | 0.806 | 0.802 |
| *Y*–8–F | 1409 | 3035 | 3010 | 2990 | 1.007 | 1.014 | 0.993 | 0.808 | 0.797 |
| *Y*–9–F | 1490 | 3220 | 3215 | 3195 | 1.001 | 1.007 | 0.994 | 0.806 | 0.798 |
| Average.. | ......... | .... | .... | .... | 1.000 | 1.0029 | 0.9951 | 0.8027 | 0.8020 |
| *Y*–1–½ | 786 | 1025 | 1040 | 1040 | 0.986 | 0.986 | 1.000 | 0.791 | 0.794 |
| *Y*–2–½ | 876 | 1160 | 1152 | 1152 | 1.007 | 1.007 | 1.000 | 0.797 | 0.790 |
| *Y*–3–½ | 952 | 1271 | 1267 | 1267 | 1.004 | 1.003 | 0.999 | 0.800 | 0.795 |
| *Y*–4–½ | 1026 | 1382 | 1380 | 1381 | 1.001 | 1.000 | 1.000 | 0.798 | 0.795 |
| *Y*–5–½ | 1113 | 1510 | 1503 | 1508 | 1.004 | 1.001 | 1.003 | 0.802 | 0.802 |
| *Y*–6–½ | 1195 | 1636 | 1620 | 1630 | 1.009 | 1.003 | 1.004 | 0.803 | 0.802 |
| *Y*–7–½ | 1287 | 1756 | 1740 | 1750 | 1.009 | 1.003 | 1.004 | 0.798 | 0.796 |
| *Y*–8–½ | 1370 | 1895 | 1872 | 1886 | 1.011 | 1.004 | 1.006 | 0.799 | 0.799 |
| *Y*–9–½ | 1462 | 2015 | 1990 | 2020 | 1.012 | 0.998 | 0.014 | 0.802 | 0.802 |
| Average........... | | .... | .... | .... | 1.0048 | 1.0005 | 1.0033 | 0.7990 | 0.7972 |
| Net Average...... | | .... | .... | .... | 1.0024 | 1.0017 | 0.9992 | 0.8002 | 0.7996 |

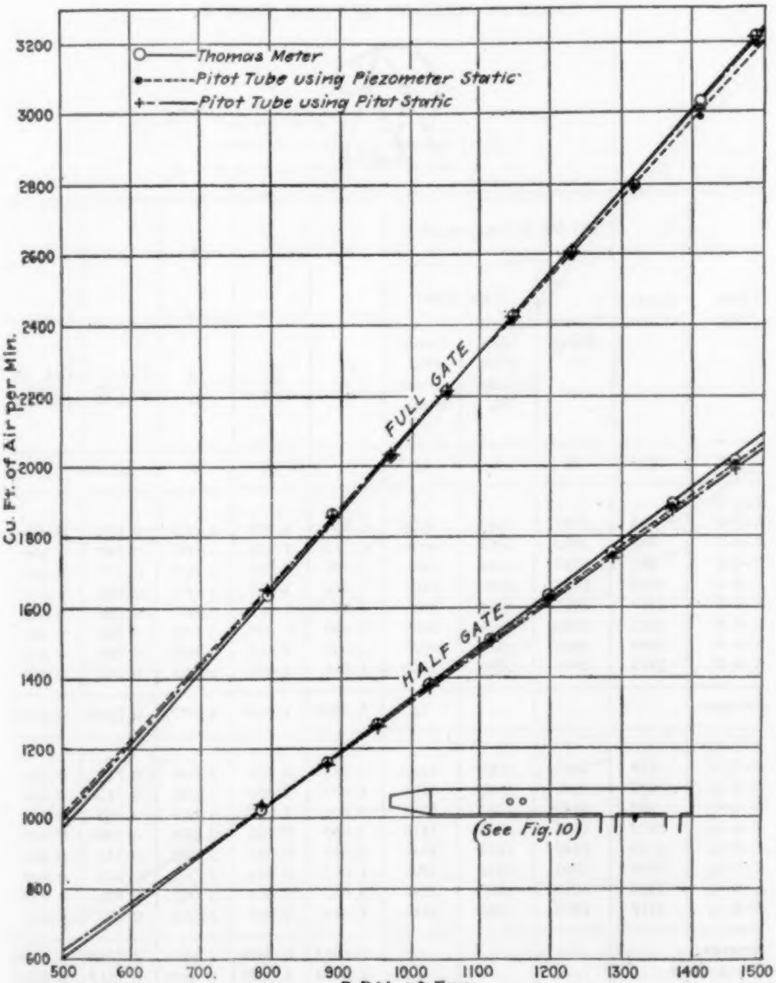



Fig. 22 Pitot Tube Y

TABLE 13 RESULTS OF PITOT TUBE $K$


| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | $M$ | $N$ | $Q$ | $U$ | $Z$ |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube | | | | | | |
| | | | Using Pitot Static $C_2$ | Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| $K$-1-F | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| $K$-2-F | 840 | 1707 | 1542 | 1673 | 1.107 | 1.020 | 1.085 | 0.773 | 0.808 |
| $K$-3-F | 910 | 1863 | 1675 | 1826 | 1.112 | 1.019 | 1.090 | 0.767 | 0.810 |
| $K$-4-F | 985 | 2020 | 1832 | 1985 | 1.102 | 1.017 | 1.083 | 0.771 | 0.813 |
| $K$-5-F | 1060 | 2186 | 2003 | 2160 | 1.091 | 1.012 | 1.077 | 0.763 | 0.818 |
| $K$-6-F | 1140 | 2363 | 2162 | 2308 | 1.091 | 1.022 | 1.067 | 0.788 | 0.822 |
| $K$-7-F | 1215 | 2527 | 2310 | 2480 | 1.093 | 1.018 | 1.073 | 0.762 | 0.802 |
| $K$-8-F | 1295 | 2695 | 2487 | 2672 | 1.083 | 1.009 | 1.073 | 0.762 | 0.818 |
| $K$-9-F | 1375 | 2850 | 2660 | 2842 | 1.070 | 1.003 | 1.068 | 0.763 | 0.797 |
| Average | ........ | .... | .... | .... | 1.0936 | 1.0150 | 1.0770 | 0.7686 | 0.8110 |
| $K$-1-½ | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| $K$-2-½ | 828 | 1077 | 1063 | 1098 | 1.015 | 0.983 | 1.033 | 0.758 | 0.808 |
| $K$-3-½ | 900 | 1192 | 1147 | 1193 | 1.039 | 1.000 | 1.038 | 0.745 | 0.805 |
| $K$-4-½ | 982 | 1307 | 1240 | 1294 | 1.054 | 1.010 | 1.043 | 0.752 | 0.814 |
| $K$-5-½ | 1070 | 1405 | 1338 | 1413 | 1.050 | 0.995 | 1.054 | 0.758 | 0.815 |
| $K$-6-½ | 1140 | 1498 | 1441 | 1510 | 1.040 | 0.993 | 1.047 | 0.752 | 0.806 |
| $K$-7-½ | 1216 | 1581 | 1518 | 1595 | 1.042 | 0.992 | 1.050 | 0.752 | 0.812 |
| $K$-8-½ | 1295 | 1675 | 1620 | 1728 | 1.033 | 0.971 | 1.067 | 0.760 | 0.819 |
| $K$-9-½ | 1377 | 1802 | 1720 | 1812 | 1.048 | 0.995 | 1.053 | 0.761 | 0.817 |
| Average | .......... | .... | .... | .... | 1.0401 | 0.9924 | 1.0481 | 0.7548 | 0.8120 |
| Net Average | ...... | .... | .... | .... | 1.0669 | 1.0037 | 1.0626 | 0.7617 | 0.8115 |

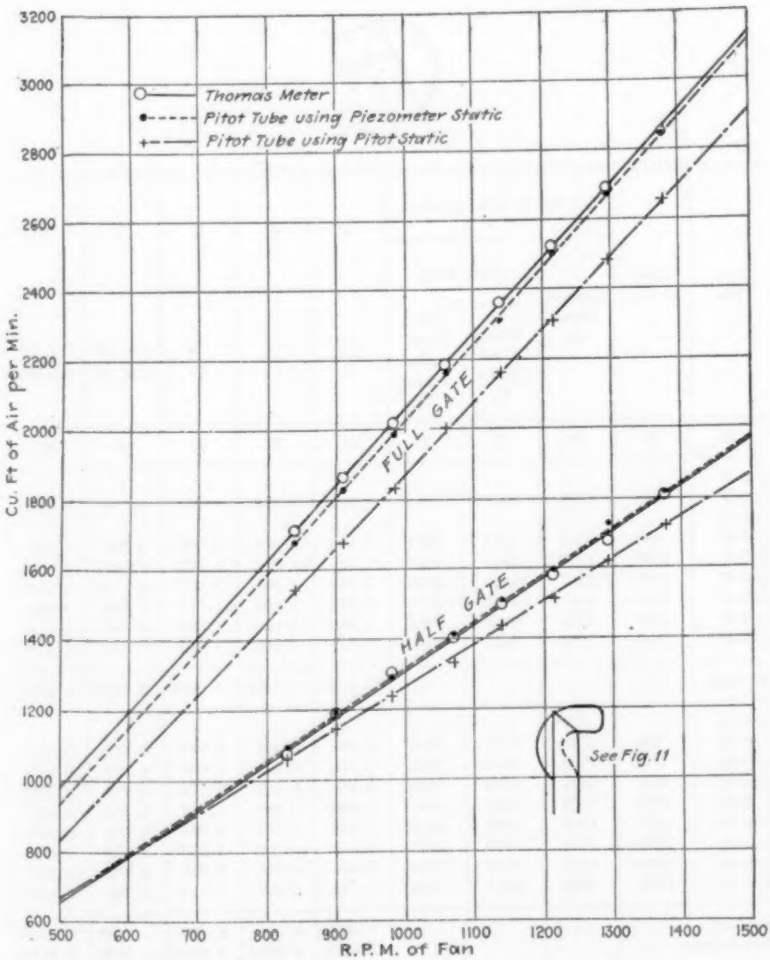



FIG. 23 PITOT TUBE *K*

TABLE 14 RESULTS OF PITOT TUBE *L*


| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | Q | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Pitot Tube: Using Pitot Static $C_2$ | Pitot Tube: Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *L*-1-F | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *L*-2-F | 838 | 1733 | 1678 | 1684 | 1.032 | 1.028 | 1.003 | 0.758 | 0.815 |
| *L*-3-F | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *L*-4-F | 980 | 2010 | 1980 | 1993 | 1.014 | 1.007 | 1.006 | 0.757 | 0.798 |
| *L*-5-F | 1060 | 2200 | 2145 | 2165 | 1.025 | 1.015 | 1.009 | 0.748 | 0.806 |
| *L*-6-F | 1140 | 2357 | 2318 | 2335 | 1.016 | 1.008 | 1.007 | 0.753 | 0.810 |
| *L*-7-F | 1210 | 2517 | 2470 | 2492 | 1.017 | 1.009 | 1.007 | 0.737 | 0.802 |
| *L*-8-F | 1295 | 2700 | 2672 | 2685 | 1.010 | 1.004 | 1.004 | 0.757 | 0.809 |
| *L*-9-F | 1365 | 2878 | 2833 | 2870 | 1.015 | 1.002 | 1.013 | 0.752 | 0.812 |
| Average | ........ | .... | .... | .... | 1.0178 | 1.0104 | 1.0068 | 0.7517 | 0.8074 |
| *L*-1-½ | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *L*-2-½ | 828 | 1100 | 1120 | 1105 | 0.982 | 0.996 | 0.987 | 0.775 | 0.805 |
| *L*-3-½ | 900 | 1209 | 1208 | 1213 | 1.000 | 0.997 | 1.002 | 0.769 | 0.816 |
| *L*-4-½ | 970 | 1296 | 1300 | 1306 | 0.997 | 0.993 | 1.004 | 0.772 | 0.817 |
| *L*-5-½ | 1055 | 1402 | 1402 | 1418 | 1.000 | 0.989 | 1.010 | 0.762 | 0.807 |
| *L*-6-½ | 1127 | 1522 | 1500 | 1519 | 1.013 | 1.002 | 1.011 | 0.765 | 0.820 |
| *L*-7-½ | 1210 | 1628 | 1609 | 1625 | 1.013 | 1.002 | 1.010 | 0.772 | 0.822 |
| *L*-8-½ | 1290 | 1721 | 1710 | 1734 | 1.006 | 0.993 | 1.013 | 0.758 | 0.810 |
| *L*-9-½ | 1360 | 1838 | 1815 | 1845 | 1.012 | 0.997 | 1.016 | 0.761 | 0.822 |
| Average.......... | | .... | .... | .... | 1.0030 | 0.996 | 1.0060 | 0.7668 | 0.8150 |
| Net Average...... | | .... | .... | .... | 1.0104 | 1.0032 | 1.0064 | 0.7593 | 0.8112 |

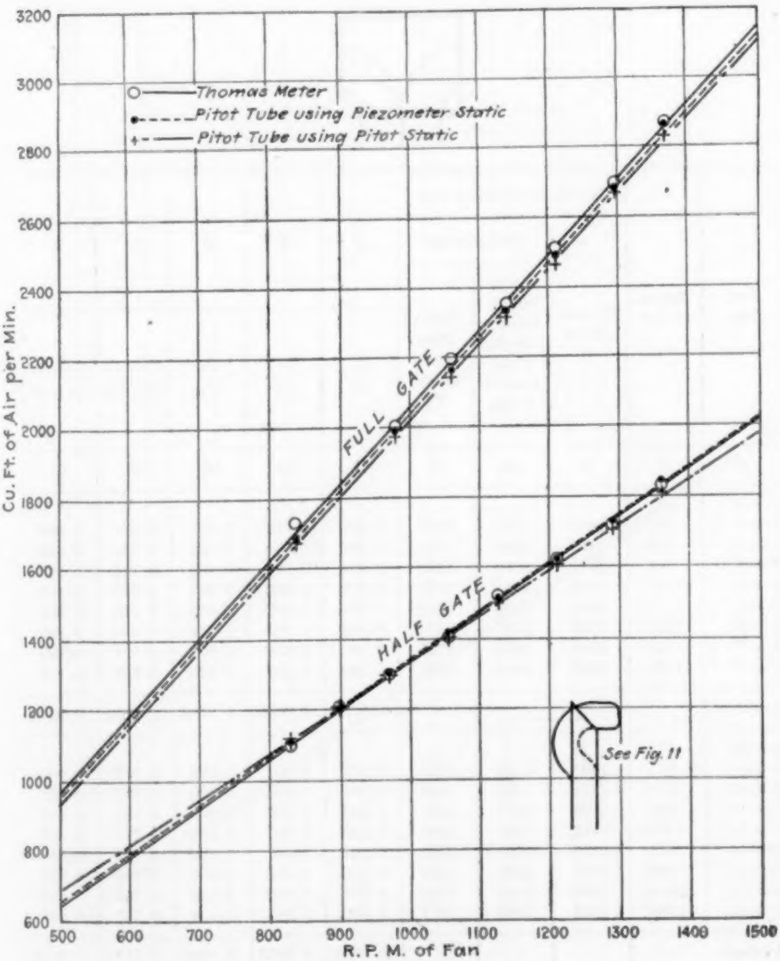
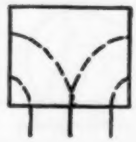



FIG. 24 PITOT TUBE *L*

TABLE 15 RESULTS OF STAUSCHEIBE *S*



| Test No. | R.p.m. of Fan | Cu. Ft. of Air per Min. | | | M | N | O | U | Z |
|---|---|---|---|---|---|---|---|---|---|
| | | Thomas Meter $C_1$ | Stauscheibe: Using Stauscheibe Static Constant—0.854 $C_2$ | Stauscheibe: Using Piezometer Static $C_3$ | $\frac{C_1}{C_2}$ | $\frac{C_1}{C_3}$ | $\frac{C_3}{C_2}$ | $\frac{\text{Col. 18}}{\text{Col. 20}}$ | $\frac{\text{Col. 19}}{\text{Col. 21}}$ |
| 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 |
| *S*-1-F | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *S*-2-F | 841 | 1695 | 1730 | 1678 | 0.981 | 1.009 | 0.971 | 0.773 | 0.807 |
| *S*-3-F | 912 | 1858 | 1900 | 1832 | 0.979 | 1.013 | 0.965 | 0.779 | 0.805 |
| *S*-4-F | 983 | 2003 | 2060 | 1986 | 0.974 | 1.008 | 0.965 | 0.774 | 0.797 |
| *S*-5-F | 1070 | 2190 | 2245 | 2182 | 0.976 | 1.003 | 0.974 | 0.778 | 0.812 |
| *S*-6-F | 1152 | 2355 | 2420 | 2353 | 0.975 | 1.001 | 0.973 | 0.782 | 0.809 |
| *S*-7-F | 1220 | 2555 | 2590 | 2522 | 0.986 | 1.012 | 0.975 | 0.790 | 0.820 |
| *S*-8-F | 1312 | 2740 | 2800 | 2722 | 0.980 | 1.005 | 0.973 | 0.797 | 0.823 |
| *S*-9-F | 1400 | 2937 | 2990 | 2900 | 0.982 | 1.012 | 0.970 | 0.808 | 0.823 |
| Average | ........ | .... | .... | .... | 0.9791 | 1.0079 | 0.9708 | 0.7851 | 0.8245 |
| *S*-1-½ | .... | .... | .... | .... | ..... | ..... | ..... | ..... | ..... |
| *S*-2-½ | 841 | 1093 | 1120 | 1092 | 0.978 | 1.001 | 0.978 | 0.772 | 0.788 |
| *S*-3-½ | 918 | 1214 | 1223 | 1208 | 0.995 | 1.004 | 0.990 | 0.785 | 0.831 |
| *S*-4-½ | 997 | 1320 | 1311 | 1320 | 1.005 | 1.000 | 1.005 | 0.767 | 0.840 |
| *S*-5-½ | 1074 | 1417 | 1427 | 1423 | 0.993 | 0.997 | 0.997 | 0.777 | 0.832 |
| *S*-6-½ | 1152 | 1505 | 1520 | 1520 | 0.992 | 0.991 | 1.000 | 0.772 | 0.824 |
| *S*-7-½ | 1232 | 1629 | 1621 | 1625 | 1.003 | 1.002 | 1.002 | 0.770 | 0.817 |
| *S*-8-½ | 1314 | 1720 | 1743 | 1750 | 0.988 | 0.985 | 1.004 | 0.768 | 0.815 |
| *S*-9-½ | 1399 | 1843 | 1860 | 1877 | 0.990 | 0.982 | 1.008 | 0.772 | 0.821 |
| Average.......... | | .... | .... | .... | 0.9930 | 0.9952 | 0.9980 | 0.7729 | 0.8210 |
| Net Average...... | | .... | .... | .... | 0.9861 | 1.0016 | 0.9844 | 0.7780 | 0.8228 |

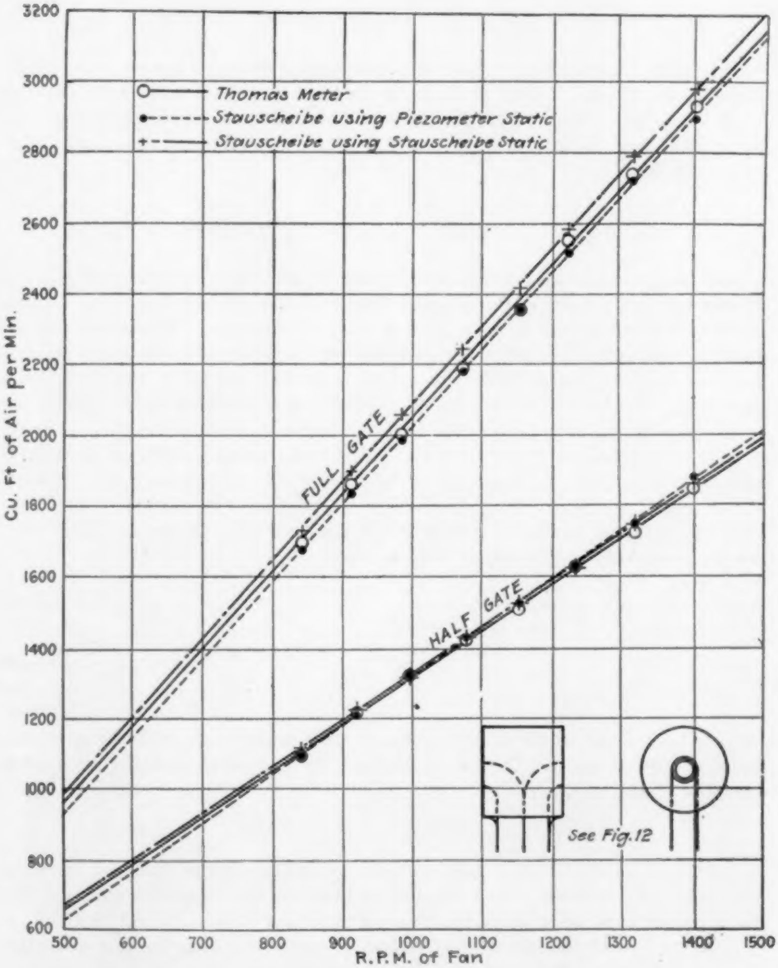



FIG. 25 STAUSCHEIBE *S*

# APPENDIX NO. 1

## THE ACCURACY OF THE THOMAS METER

49 The Thomas meter has been thoroughly tested under various conditions of service and a few of the tests are briefly referred to in the following notes in order to show that it is a reliable and accurate means for measuring gases and that its use as a standard meter in these experiments with pitot tubes was justified.

### TESTS AGAINST CALIBRATED PITOT TUBES[1]

50 A Thomas meter was tested by the People's Natural Gas Company of Pittsburgh at their Brave Pumping Station, in a 10-in. suction line from the gas wells to the pump and in series with a pitot tube station. This latter was located a mile and a half from the pumping station where the electric meter was installed and every precaution was taken to prevent leakage in the intervening pipe line. This pitot tube station was developed after years of experiment and at great expense. The pitot tubes in the station were carefully calibrated under working conditions and were known to give accurate results when the calibration constants were used. A 45-day test was run, from April 7 to June 3, 1911, during which period the rate of flow varied from 90,000 to 640,000 cu. ft. per hour, the pressure of the gas varied from 46 to 185 lb. gage and the temperature varied from 45 to 65 deg. fahr. The results were as follows:

| | |
|---|---|
| Total standard cu. ft. of gas by pitot tube | 337,546,182 |
| Total standard cu. ft. of gas by Thomas meter | 336,732,018 |
| Difference | 814,164 |
| Per cent of difference | 0.24 |

A year later a test of the same meter was made without any changes of adjustment whatever and the Thomas meter was found to agree with the pitot tubes within 0.42 per cent.

### TESTS AGAINST HOLDERS[1]

51 A Thomas meter is used to measure the gas in the common discharge line from the booster station of the Milwaukee Gas Light Company. The holder tests were made with the greatest care on October 13 and 14, 1911, at a time when the temperature had remained constant during the day and night for several days. Two different holders were used. One test was made at the maximum capacity of the meter when the gas was being pumped from the large holder through the meter at a pressure of about 40 in. of water. A second test was made at the minimum capacity of the meter when gas was flowing from smaller holder through the meter at holder pressure.

[1] Carl C. Thomas. Proc. Am. Gas Inst., vol. 7, 1912.

52 The results were as follows:

| Holder used | Larger | Smaller |
|---|---|---|
| Duration of test | 2 hr. 15 min. | 5 hr. |
| Total cu. ft. of air by Thomas meter at 30 in. mercury and 60 deg. fahr. | 1,956,000 | 497,500 |
| Total cu. ft. of air by holder at 30 in. mercury and 60 deg. fahr. | 1,958,691 | 498,628 |
| Difference | 2,691 | 1,128 |
| Per cent of difference | 0.137 | 0.224 |

### TESTS AGAINST WET METERS[1]

53 A Thomas meter was imported into Germany to be used in some scientific investigations of blowers and compressors. It was tested against a carefully calibrated wet meter at the Berlin IV Precinct Gas Works under the direct supervision of their engineers, who are of recognized technical ability and who exercised the most painstaking care throughout the tests. A few representative results of the Berlin tests are given below:

| | | | |
|---|---|---|---|
| Duration of test, hours | 1½ | 3½ | 3½ |
| Cubic meters of gas by wet meter at 15.5 deg. cent. and 760 | 5410 | 7111 | 14600 |
| Cubic meters of gas by Thomas meter | 5400 | 7076 | 14537 |
| Difference | 10 | 35 | 63 |
| Per cent difference | 0.20 | 0.49 | 0.46 |

### TESTS AT THE WORKS OF THE CUTLER-HAMMER MANUFACTURING COMPANY

54 Experimental work is constantly being carried on by the manufacturers of the Thomas meter. Two of the most interesting tests conducted there are described here through the courtesy of the Cutler-Hammer Manufacturing Company, as they demonstrate the accuracy of the Thomas meter under varying conditions.

55 *First Test.* Two automatically operated Thomas meters, one of 25,000 cu. ft. per hour capacity and the other of 50,000 cu. ft. per hour capacity, were put in the same pipe line in series. When air was flowing through the pipe it was noted that each meter recorded the same amount of air. By means of an electric heater in the pipe between the two meters the temperature of the air entering the second meter was gradually increased until it was 60 deg. fahr. higher than the temperature of the air entering the first meter. The amounts recorded on each meter were meanwhile carefully watched and it was observed that the readings still remained practically identical.

56 *Second Test.* A manually operated test meter was connected in a *horizontal* position in series with an automatically controlled meter which was in a vertical position, with a right angled bend intervening between them. This arrangement was made purposely to prevent the air passing through the two meters in the same manner. The test meter was manually operated on both 110 and 220 volts direct current and the automatic meter on 220 volts alternating current. The automatic meter had a capacity of 500,000 cu. ft. of free gas per hour. The results at different rates of flow were as follows:

| | |
|---|---|
| 9 per cent of maximum flow, error in automatic meter | +0.2 per cent |
| 42 per cent of maximum flow, error in automatic meter | +0.2 per cent |
| 81 per cent of maximum flow, error in automatic meter | +0.0 per cent |

[1] Carl C. Thomas. Proc. Am. Gas Inst., vol. 7, 1912.

# APPENDIX NO. 2

## DATA AND CALCULATIONS

57 In order to show the manner in which the results of the calculations upon the pitot tubes were tabulated, Tables 16 to 18, applying to tubes *A*, *H* and *X* are here reproduced. The explanation of the calculations is as follows:

58 In column 1, Test Number, the first letter designates the pitot tube; the middle figure, the test in the series arranged according to the fan speed; and the last symbol, *F* or ½, signifies full gate or half gate at the discharge end of the test pipe.

59 Column 2 gives the average revolutions per minute of the fan as obtained by a hand revolution counter.

60 Column 3 shows the barometer reading in inches of mercury.

61 In column 4, the atmospheric pressure in pounds per sq. in. is obtained by multiplying the values in column 3 by the weight of a cu. in. of mercury taken from Chart *F* corresponding to room temperature.

62 Column 5 indicates the pressure in the pipe above atmospheric pressure at the point where the pitot tube was inserted.

63 In column 6, the pressure inside the pipe above the atmospheric pressure is the product of the values given in column 5, the specific gravity of the gasolene and the weight of the water per cu. in. taken from Chart *G* corresponding to room temperature.

64 In column 7, the absolute pressure in lb. per sq. in. on the air flowing through pipe at the pitot tube is the sum of the pressures given in column 4 and column 6.

65 Columns 8, 9 and 10 give the averages of the observed temperatures in deg. fahr.

66 In column 11, the percentage of humidity of the flowing air is obtained from Chart *A* and the temperatures given in columns 9 and 10.

67 In column 12, the weight of air partially saturated with water vapor in lb. per cu. ft. is obtained from Charts *D* and *E*. Knowing the temperature of the air (column 10) and the percentage of humidity (column 11) the weight of a cubic foot of air at 14 lb. per sq. in. pressure may be read from Chart *D*. To this value add the correction obtained from Chart *E* corresponding to the pressures given in column 7 and the temperature in column 10. The sum is the weight of a cubic foot of the mixture.

68 The calculations involved in finding the weight of a cubic foot of air partially saturated with water vapor are outlined in Appendix No. 3 and are based on well-known thermodynamic relations.

69 In column 13, the specific heat of the mixture of air and water vapor in B.t.u. per lb. is obtained from Chart *B* corresponding to the temperatures given in column 10 and the percentage of humidity given in column 11. The method of calculating specific heat of a mixture of air and water vapor for any temperature, pressure and per cent humidity is shown in Appendix No. 3.

TABLE 16 TABULATED CALCULATIONS, PITOT TUBE *A*

| | | Pressures | | | | | Temperatures | | | | | | Thomas Meter | | | | Pitot Tube | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Barometer | | Static | | | | | | | | | | | | | Velocity Heads | | | | | | Cu. Ft. of Air per Min. | |
| | | | | | | | | | | | | | | | | | In. Gasolene, Sp. Gr. 0.733 | | | | Ft. of Air | | | |
| Test No. | R.p.m. of Fan | In. of Mercury | Lb. per Sq. In. | In. of Gasolene, Sp. Gr. | Lb. per Sq. In. | Absolute Lb. per Sq. In. | Room, Deg. Fahr. | Wet Bulb, Deg. Fahr. | Dry Bulb, Deg. Fahr. | Humidity, per cent | Weight of Air, Lb. per Cu. Ft. | Specific Heat of Air B.t.u. per Lb. | Calibration Constant, $K$ | Amperes Corrected | Volts Corrected | Cu. Ft. of Air per Min., $C_1$ | Average Using Pitot Static, $H_1$ | Average Using Piezometer Static, $H_2$ | Center Using Pitot Static, $H_3$ | Center Using Piezometer Static, $H_4$ | Average Using Pitot Static, $h_1$ | Average Using Piezometer Static, $h_2$ | Using Pitot Static $C_2$ | Using Piezometer Static, $C_3$ |
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
| *A*-1-½ | 794 | 29.200 | 14.267 | 1.17 | 0.031 | 14.298 | 80.0 | 69.8 | 84.0 | 48.5 | 0.07049 | 0.2445 | 0.02835 | 12.49 | 52.7 | 1080 | 0.1336 | 0.1434 | 0.1725 | 0.1765 | 7.22 | 7.75 | 1030 | 1068 |
| *A*-2-½ | 875 | 29.206 | 14.270 | 1.37 | 0.036 | 14.307 | 81.5 | 70.9 | 85.1 | 49.0 | 0.07036 | 0.2446 | 0.02839 | 12.98 | 55.15 | 1178 | 0.1629 | 0.1723 | 0.2110 | 0.2170 | 8.82 | 9.34 | 1138 | 1171 |
| *A*-3-½ | 951 | 29.210 | 14.272 | 1.41 | 0.037 | 14.309 | 82.2 | 71.4 | 86.3 | 48.5 | 0.07021 | 0.2446 | 0.02843 | 13.73 | 57.8 | 1315 | 0.1928 | 0.2060 | 0.2500 | 0.2650 | 10.47 | 11.18 | 1240 | 1280 |
| *A*-4-½ | 1028 | 29.210 | 14.272 | 1.69 | 0.045 | 14.317 | 83.1 | 71.8 | 87.1 | 48.0 | 0.07013 | 0.2447 | 0.02848 | 14.33 | 60.05 | 1420 | 0.2355 | 0.2450 | 0.2970 | 0.3095 | 12.80 | 13.32 | 1371 | 1399 |
| *A*-5-½ | 1107 | 29.212 | 14.273 | 1.96 | 0.052 | 14.325 | 84.0 | 72.9 | 88.4 | 48.0 | 0.06999 | 0.2448 | 0.02853 | 14.88 | 62.75 | 1552 | 0.2645 | 0.2860 | 0.3520 | 0.3670 | 14.40 | 15.57 | 1452 | 1510 |
| *A*-6-½ | 1190 | 29.212 | 14.273 | 2.28 | 0.060 | 14.333 | 85.7 | 74.2 | 90.7 | 46.5 | 0.06969 | 0.2450 | 0.02865 | 15.43 | 64.75 | 1673 | 0.3087 | 0.3440 | 0.3980 | 0.4245 | 16.88 | 18.81 | 1573 | 1662 |
| *A*-7-½ | 1266 | 29.210 | 14.272 | 2.60 | 0.069 | 14.341 | 86.8 | 74.8 | 91.7 | 46.0 | 0.06961 | 0.2451 | 0.02871 | 15.99 | 67.05 | 1803 | 0.3447 | 0.3893 | 0.4510 | 0.4880 | 18.87 | 21.31 | 1663 | 1770 |
| *A*-8-½ | 1371 | 29.210 | 14.272 | 3.00 | 0.079 | 14.351 | 87.8 | 75.0 | 93.1 | 43.5 | 0.06949 | 0.2452 | 0.02878 | 16.49 | 69.75 | 1942 | 0.3956 | 0.4495 | 0.5150 | 0.5585 | 21.68 | 24.65 | 1782 | 1902 |
| *A*-9-½ | 1447 | 29.208 | 14.271 | 3.38 | 0.089 | 14.361 | 88.5 | 75.2 | 94.0 | 41.5 | 0.06945 | 0.2452 | 0.02883 | 16.99 | 71.20 | 2049 | 0.4450 | 0.5122 | 0.5860 | 0.6360 | 24.40 | 28.10 | 1892 | 2030 |
| *A*-1-F | 795 | 29.142 | 14.239 | 0.18 | 0.005 | 14.244 | 86.0 | 77.2 | 89.2 | 58.0 | 0.06933 | 0.2456 | 0.02860 | 15.52 | 65.7 | 1710 | 0.3392 | 0.3508 | 0.4380 | 0.4440 | 18.64 | 19.28 | 1655 | 1680 |
| *A*-2-F | 870 | 29.150 | 14.243 | 0.22 | 0.006 | 14.249 | 86.5 | 76.9 | 89.8 | 56.0 | 0.06928 | 0.2455 | 0.02862 | 16.37 | 69.4 | 1910 | 0.4135 | 0.4335 | 0.5405 | 0.5505 | 22.74 | 23.85 | 1828 | 1870 |
| *A*-3-F | 947 | 29.148 | 14.242 | 0.27 | 0.007 | 14.249 | 87.0 | 77.1 | 90.5 | 54.5 | 0.06923 | 0.2455 | 0.02865 | 17.07 | 72.6 | 2085 | 0.4935 | 0.5235 | 0.6430 | 0.6635 | 27.15 | 28.80 | 1995 | 2058 |
| *A*-4-F | 1031 | 29.150 | 14.243 | 0.32 | 0.008 | 14.251 | 88.0 | 77.2 | 91.2 | 53.5 | 0.06912 | 0.2456 | 0.02870 | 17.72 | 75.7 | 2265 | 0.5780 | 0.6180 | 0.7520 | 0.7880 | 31.85 | 34.06 | 2160 | 2237 |
| *A*-5-F | 1118 | 29.148 | 14.242 | 0.37 | 0.010 | 14.252 | 88.8 | 78.0 | 92.8 | 52.0 | 0.06890 | 0.2457 | 0.02877 | 18.67 | 79.5 | 2487 | 0.6770 | 0.7365 | 0.8810 | 0.9310 | 37.43 | 40.73 | 2340 | 2440 |
| *A*-6-F | 1191 | 29.148 | 14.242 | 0.41 | 0.011 | 14.253 | 89.7 | 78.5 | 93.4 | 51.5 | 0.06882 | 0.2458 | 0.02879 | 19.22 | 81.9 | 2675 | 0.7780 | 0.8480 | 1.0040 | 1.0695 | 43.05 | 46.93 | 2513 | 2620 |
| *A*-7-F | 1273 | 29.150 | 14.243 | 0.46 | 0.012 | 14.255 | 90.0 | 78.9 | 94.3 | 51.0 | 0.06872 | 0.2459 | 0.02885 | 19.97 | 84.6 | 2885 | 0.8865 | 0.9685 | 1.1465 | 1.2290 | 49.16 | 53.65 | 2685 | 2805 |
| *A*-8-F | 1370 | 29.148 | 14.243 | 0.54 | 0.014 | 14.256 | 90.0 | 79.0 | 94.8 | 50.0 | 0.06867 | 0.2459 | 0.02887 | 20.67 | 87.4 | 3095 | 1.0220 | 1.1110 | 1.3100 | 1.4135 | 56.70 | 61.65 | 2880 | 3005 |
| *A*-9-F | 1458 | 29.130 | 14.233 | 0.62 | 0.016 | 14.249 | 91.0 | 79.6 | 96.3 | 48.5 | 0.06847 | 0.2458 | 0.02893 | 21.27 | 90.2 | 3300 | 1.1640 | 1.2815 | 1.4790 | 1.6070 | 64.75 | 71.30 | 3080 | 3240 |

70 In column 14, the value of $K$ depends upon the calibration of the resistance thermometers in the Thomas meter and is taken from the curve given in Fig. 26. It is defined by its use in the following formula, referred to later under column 17:

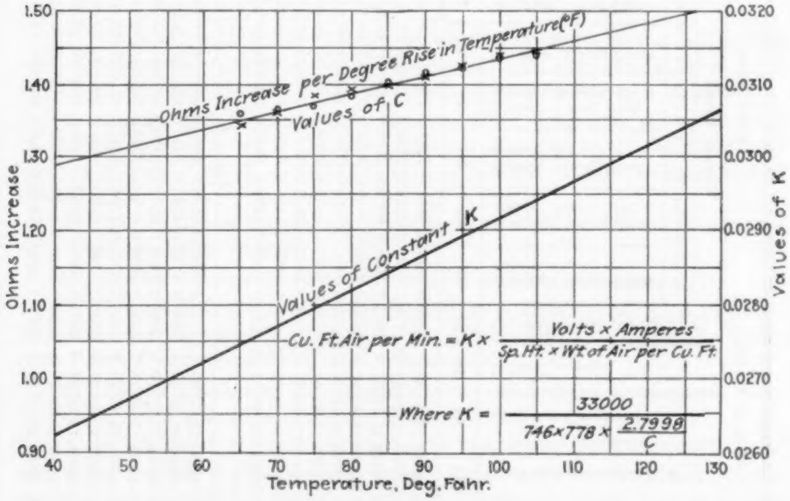



Fig. 26 Calibration of Resistance Thermometers in Thomas Electric Meter

$$\text{Cu. ft. of air per minute by the Thomas meter} = K \frac{\text{(volts) (amperes)}}{\text{(specific heat) (wt. air per cu. ft.)}}$$

where

$$K = \frac{33000}{746 \times 778 \times \frac{2.7998}{C}}$$

See also the description of the Thomas meter in Pars. 13 to 17.

71 Columns 15 and 16 show the corrected readings of the voltmeter and ammeter which measured the electrical energy consumed by the heater in the Thomas meter.

72 In column 17, the cubic feet of air per minute as measured by the Thomas meter is calculated from the following formula:

$$C_1 = K \frac{(V)\,(A)}{SW}$$

where

$C_1$ = cu. ft. of air per min. as measured by the Thomas meter

TABLE 17 TABULATED CALCULATIONS, PITOT TUBE *X*

| Test No. | R.p.m. of Fan | Pressures: Barometer: In. of Mercury | Pressures: Barometer: Lb. per Sq. In. | Pressures: Static: In. of Gasolene, Sp. Gr. | Pressures: Static: Lb. per Sq. In. | Pressures: Absolute Lb. per Sq. In. | Temperatures: Room, Deg. Fahr. | Temperatures: Wet Bulb, Deg. Fahr. | Temperatures: Dry Bulb, Deg. Fahr. | Humidity, per Cent | Weight of Air, Lb. per Cu. Ft. | Specific Heat of Air B.t.u. per Lb. | Thomas Meter: Calibration Constant, *K* | Thomas Meter: Amperes Corrected | Thomas Meter: Volts Corrected | Thomas Meter: Cu. Ft. of Air per Min., $C_1$ | Pitot Tube: Velocity Heads: In. Gasolene, Sp.Gr. 0.740–43: Average Using Pitot Static, $H_1$ | Pitot Tube: Velocity Heads: In. Gasolene: Average Using Piezometer Static, $H_2$ | Pitot Tube: Velocity Heads: In. Gasolene: Center Using Pitot Static, $H_3$ | Pitot Tube: Velocity Heads: In. Gasolene: Center Using Piezometer Static, $H_4$ | Pitot Tube: Velocity Heads: Ft. of Air: Average Using Pitot Static, $h_1$ | Pitot Tube: Velocity Heads: Ft. of Air: Average Using Piezometer Static, $h_2$ | Pitot Tube: Cu. Ft. of Air per Min.: Using Pitot Static $C_3$ | Pitot Tube: Cu. Ft. of Air per Min.: Using Piezometer Static, $C_4$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
| *X*-1-½ | 785 | 29.100 | 14.200 | 0.90 | 0.024 | 14.224 | 95.0 | 83.0 | 98.5 | 52.5 | 0.06792 | 0.2465 | 0.02905 | 11.94 | 50.51 | 1047 | 0.1102 | 0.1293 | 0.1495 | 0.1685 | 6.25 | 7.33 | 958 | 1037 |
| *X*-2-½ | 876 | 29.078 | 14.185 | 1.12 | 0.030 | 14.215 | 95.8 | 81.8 | 99.0 | 48.0 | 0.06788 | 0.2463 | 0.02907 | 12.51 | 53.32 | 1161 | 0.1348 | 0.1596 | 0.1795 | 0.2025 | 7.66 | 9.07 | 1060 | 1153 |
| *X*-3-½ | 954 | 29.058 | 14.177 | 1.35 | 0.036 | 14.213 | 96.0 | 81.5 | 99.0 | 47.5 | 0.06787 | 0.2462 | 0.02907 | 13.16 | 55.83 | 1270 | 0.1603 | 0.1901 | 0.2115 | 0.2410 | 9.10 | 10.80 | 1156 | 1260 |
| *X*-4-½ | 1028 | 29.016 | 14.165 | 1.60 | 0.042 | 14.207 | 96.0 | 80.9 | 99.1 | 46.0 | 0.06785 | 0.2461 | 0.02908 | 13.68 | 57.92 | 1378 | 0.1896 | 0.2287 | 0.2510 | 0.2940 | 10.75 | 12.98 | 1256 | 1380 |
| *X*-5-½ | 1120 | 29.028 | 14.170 | 1.86 | 0.049 | 14.219 | 95.5 | 80.9 | 99.7 | 44.0 | 0.06785 | 0.2461 | 0.02911 | 14.23 | 60.02 | 1490 | 0.2230 | 0.2659 | 0.2950 | 0.3460 | 12.66 | 15.08 | 1363 | 1488 |
| *X*-6-½ | 1195 | 29.054 | 14.177 | 2.16 | 0.057 | 14.234 | 96.0 | 81.6 | 100.6 | 44.5 | 0.06780 | 0.2462 | 0.02915 | 14.73 | 62.05 | 1595 | 0.2548 | 0.3106 | 0.3360 | 0.3930 | 14.46 | 17.63 | 1457 | 1609 |
| *X*-7-½ | 1288 | 29.066 | 14.180 | 2.55 | 0.068 | 14.248 | 96.3 | 81.8 | 100.9 | 44.5 | 0.06785 | 0.2463 | 0.02917 | 15.23 | 64.45 | 1713 | 0.2918 | 0.3568 | 0.3840 | 0.4570 | 16.55 | 20.25 | 1557 | 1723 |
| *X*-8-½ | 1370 | 29.066 | 14.180 | 2.92 | 0.077 | 14.257 | 96.3 | 81.8 | 101.0 | 44.5 | 0.06786 | 0.2463 | 0.02918 | 15.79 | 66.58 | 1834 | 0.3354 | 0.4192 | 0.4380 | 0.5305 | 19.03 | 23.80 | 1670 | 1870 |
| *X*-9-½ | 1460 | 29.070 | 14.181 | 3.30 | 0.088 | 14.269 | 96.5 | 82.0 | 101.5 | 44.0 | 0.06785 | 0.2463 | 0.02920 | 16.31 | 68.72 | 1960 | 0.3792 | 0.4788 | 0.4940 | 0.6035 | 21.50 | 27.20 | 1775 | 1996 |
| *X*-1-F | 807 | 29.034 | 14.173 | 0.18 | 0.005 | 14.178 | 80.2 | 73.3 | 83.6 | 62.2 | 0.06977 | 0.2452 | 0.02834 | 15.03 | 64.55 | 1605 | 0.2840 | 0.3395 | 0.3565 | 0.4295 | 15.55 | 18.58 | 1510 | 1650 |
| *X*-2-F | 888 | 29.026 | 14.168 | 0.22 | 0.006 | 14.174 | 80.5 | 74.0 | 84.1 | 62.2 | 0.06967 | 0.2453 | 0.02835 | 15.80 | 68.75 | 1800 | 0.3404 | 0.4142 | 0.4290 | 0.5220 | 18.70 | 22.72 | 1655 | 1825 |
| *X*-3-F | 963 | 29.015 | 14.160 | 0.26 | 0.007 | 14.167 | 81.0 | 75.0 | 85.0 | 63.0 | 0.06950 | 0.2453 | 0.02838 | 16.64 | 70.80 | 1962 | 0.4049 | 0.4968 | 0.5100 | 0.6240 | 22.25 | 27.30 | 1805 | 2000 |
| *X*-4-F | 1046 | 29.014 | 14.160 | 0.31 | 0.008 | 14.168 | 81.5 | 75.4 | 85.7 | 63.0 | 0.06940 | 0.2454 | 0.02842 | 17.35 | 73.82 | 2140 | 0.4875 | 0.5874 | 0.6155 | 0.7455 | 26.65 | 32.35 | 1975 | 2180 |
| *X*-5-F | 1135 | 29.017 | 14.161 | 0.38 | 0.010 | 14.171 | 82.0 | 76.0 | 86.2 | 63.0 | 0.06935 | 0.2455 | 0.02845 | 18.15 | 77.38 | 2345 | 0.5585 | 0.7014 | 0.7270 | 0.8840 | 30.75 | 38.60 | 2125 | 2380 |
| *X*-6-F | 1229 | 29.015 | 14.160 | 0.40 | 0.011 | 14.171 | 82.5 | 76.2 | 86.9 | 62.7 | 0.06926 | 0.2456 | 0.02848 | 18.96 | 80.20 | 2545 | 0.6433 | 0.8124 | 0.8320 | 1.0250 | 35.50 | 44.80 | 2280 | 2562 |
| *X*-7-F | 1311 | 29.007 | 14.156 | 0.47 | 0.013 | 14.169 | 83.3 | 76.7 | 88.0 | 60.5 | 0.06912 | 0.2457 | 0.02853 | 19.65 | 83.27 | 2745 | 0.7451 | 0.9279 | 0.9650 | 1.1920 | 41.20 | 51.25 | 2455 | 2740 |
| *X*-8-F | 1410 | 29.006 | 14.155 | 0.52 | 0.014 | 14.169 | 84.0 | 77.3 | 89.0 | 59.5 | 0.06897 | 0.2457 | 0.02857 | 20.32 | 85.90 | 2940 | 0.8496 | 1.0863 | 1.0900 | 1.3590 | 47.05 | 60.20 | 2625 | 2970 |
| *X*-9-F | 1496 | 29.004 | 14.154 | 0.60 | 0.016 | 14.170 | 84.8 | 78.0 | 90.1 | 58.5 | 0.06885 | 0.2458 | 0.02862 | 21.01 | 88.70 | 3145 | 0.9560 | 1.2331 | 1.2290 | 1.5430 | 53.00 | 68.45 | 2785 | 3165 |

$(V)\ (A)$ = volts (column 15) times amperes (column 16) = watts consumed by electrical heater in the Thomas meter

$S$ = specific heat of the air flowing (column 13)

$W$ = weight of the air flowing in lb. per cu. ft. (column 12)

$K$ is taken from column 14

73 Column 18 gives the mean velocity head as measured by the pitot tube using the pitot tube static pressure. It was obtained as follows: Readings were taken of the velocity head at 20 points on the cross-section of the pipe as described in Par. 21 and by Fig. 6. The square roots of these 20 readings were averaged and the square of this average is the value entered in column 18.

74 Column 19 indicates the mean velocity head as measured by the pitot dynamic tube and the piezometer static pressure. It was obtained in the manner described above for column 18.

75 Columns 20 and 21 record the velocity heads when the pitot tube is at the center of the pipe, column 20 using the pitot tube static pressure and column 21 using the piezometer static pressure.

76 Columns 22 and 23 are the velocity heads given in columns 18 and 19 reduced to feet of air flowing. They were calculated as follows:

$$h = \frac{144 spH}{W}$$

where

$h$ = velocity head in ft. of air

$s$ = specific gravity of gasolene

$p$ = weight of water in pounds per cu. in., taken from Chart $G$

$H$ = velocity head in in. of gasolene as given in columns 18 or 19

$W$ = weight of air flowing in lb. per cu. ft. (column 12)

77 Column 24 gives the cubic feet of air per minute as measured by the pitot tube using the pitot tube static pressure.

78 Column 25 gives the cubic feet of air per minute as measured by the pitot dynamic tube and the piezometer static pressure.

79 The method of calculating the results given in columns 24 and 25 is as follows:

$$\text{for pitot tube } C_2 = 60\ A\sqrt{2gh_1} = 383\sqrt{h_1}$$

$$\text{for Stauscheibe } C_2 = 60\ A\frac{\sqrt{2gh_1}}{1.17} = 327\sqrt{2gh_1}$$

$$\text{for both } C_3 = 60\ A\sqrt{2gh_2}\ \sqrt{383}\sqrt{h_2}$$

where

$C_2$ = cu. ft. of air per minute as measured by pitot tube using pitot tube static pressure

$C_3$ = cu. ft. of air per minute as measured by pitot dynamic tube and piezometer static pressure

$A$ = area of cross-section of pipe where pitot tube was inserted = 0.7959 sq. ft.

$g$ = 32.2

$h_1$ = velocity head in ft. of air taken from column 22 and described previously

$h_2$ = velocity head in ft. of air taken from column 23 and described previously

TABLE 18 TABULATED CALCULATIONS, PITOT TUBE *H*

| Test No. | R.p.m. of Fan | Pressures, Barometer: In. of Mercury | Pressures, Barometer: Lb. per Sq. In. | Pressures, Static: In. of Gasolene, Sp. Gr. | Pressures, Static: Lb. per Sq. In. | Pressures: Absolute Lb. per Sq. In. | Temperatures: Room, Deg. Fahr. | Temperatures: Wet Bulb, Deg. Fahr. | Temperatures: Dry Bulb, Deg. Fahr. | Humidity, per Cent | Weight of Air, Lb. per Cu. Ft. | Specific Heat of Air B.t.u. per Lb. | Thomas Meter: Calibration Constant, $K$ | Thomas Meter: Amperes Corrected | Thomas Meter: Volts Corrected | Thomas Meter: Cu. Ft. of Air per Min., $C_1$ | Pitot Tube, Velocity Heads, In. Gasolene, Sp. Gr. 0.745: Average Using Pitot Static, $H_1$ | Average Using Piezometer Static, $H_2$ | Center Using Pitot Static, $H_3$ | Center Using Piezometer Static, $H_4$ | Ft. of Air: Average Using Pitot Static, $h_1$ | Ft. of Air: Average Using Piezometer Static, $h_2$ | Cu. Ft. of Air per Min.: Using Pitot Static $C_2$ | Cu. Ft. of Air per Min.: Using Piezometer Static, $C_3$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
| *H*-1-½ | .... | ...... | ...... | ...... | ...... | ...... | .... | .... | .... | .... | ...... | ...... | ...... | .... | .... | .... | ...... | ...... | ...... | ...... | .... | .... | .... | .... |
| *H*-2-½ | 870 | 29.182 | 14.270 | 1.22 | 0.032 | 14.302 | 73.5 | 63.0 | 79.0 | 40.5 | 0.07129 | 0.2437 | 0.02808 | 13.33 | 55.93 | 1203 | 0.1718 | 0.1737 | 0.2180 | 0.2190 | 9.33 | 9.43 | 1170 | 1176 |
| *H*-3-½ | 948 | 29.182 | 14.270 | 1.46 | 0.039 | 14.309 | 73.5 | 63.8 | 80.2 | 40.0 | 0.07127 | 0.2438 | 0.02814 | 13.91 | 58.28 | 1313 | 0.2039 | 0.2067 | 0.2630 | 0.2650 | 11.05 | 11.22 | 1274 | 1284 |
| *H*-4-½ | 1029 | 29.182 | 14.270 | 1.70 | 0.045 | 14.315 | 76.0 | 65.3 | 82.1 | 40.0 | 0.07092 | 0.2439 | 0.02824 | 14.46 | 60.42 | 1425 | 0.2410 | 0.2459 | 0.3065 | 0.3125 | 13.14 | 13.40 | 1390 | 1403 |
| *H*-5-½ | 1110 | 29.182 | 14.270 | 2.05 | 0.054 | 14.324 | 77.5 | 66.0 | 83.3 | 39.5 | 0.07080 | 0.2439 | 0.02830 | 15.20 | 63.15 | 1570 | 0.2900 | 0.2943 | 0.3610 | 0.3720 | 15.80 | 16.05 | 1523 | 1535 |
| *H*-6-½ | 1185 | 29.182 | 14.270 | 2.34 | 0.062 | 14.332 | 78.9 | 66.8 | 84.6 | 39.0 | 0.07066 | 0.2440 | 0.02836 | 15.66 | 65.52 | 1685 | 0.3284 | 0.3407 | 0.4155 | 0.4280 | 17.97 | 18.62 | 1623 | 1654 |
| *H*-7-½ | 1265 | 29.184 | 14.270 | 2.72 | 0.072 | 14.342 | 79.9 | 67.8 | 85.9 | 39.0 | 0.07052 | 0.2441 | 0.02844 | 16.15 | 67.42 | 1793 | 0.3774 | 0.3879 | 0.4745 | 0.4900 | 20.62 | 21.20 | 1738 | 1763 |
| *H*-8-½ | 1360 | 29.186 | 14.270 | 3.08 | 0.082 | 14.352 | 81.5 | 69.5 | 88.0 | 39.0 | 0.07027 | 0.2443 | 0.02853 | 16.71 | 69.87 | 1932 | 0.4330 | 0.4516 | 0.5440 | 0.5640 | 23.80 | 24.79 | 1870 | 1910 |
| *H*-9-½ | 1454 | 29.186 | 14.270 | 3.51 | 0.093 | 14.363 | 82.5 | 70.2 | 89.4 | 38.5 | 0.07013 | 0.2444 | 0.02860 | 17.24 | 72.15 | 2072 | 0.4881 | 0.5153 | 0.6200 | 0.6480 | 26.90 | 28.43 | 1986 | 2040 |
| *H*-1-F | .... | ...... | ...... | ...... | ...... | ...... | .... | .... | .... | .... | ...... | ...... | ...... | .... | .... | .... | ...... | ...... | ...... | ...... | .... | .... | .... | .... |
| *H*-2-F | 872 | 29.150 | 14.245 | 0.230 | 0.006 | 14.251 | 80.0 | 68.7 | 83.9 | 46.5 | 0.07029 | 0.2448 | 0.02833 | 16.74 | 70.24 | 1930 | 0.4579 | 0.4541 | 0.5700 | 0.5690 | 25.20 | 24.95 | 1922 | 1912 |
| *H*-3-F | 947 | 29.162 | 14.248 | 0.280 | 0.007 | 14.255 | 81.0 | 68.7 | 84.7 | 44.3 | 0.07021 | 0.2448 | 0.02837 | 17.47 | 73.28 | 2108 | 0.5592 | 0.5556 | 0.6735 | 0.6725 | 30.75 | 30.75 | 2123 | 2123 |
| *H*-4-F | 1024 | 29.168 | 14.252 | 0.310 | 0.008 | 14.260 | 82.1 | 69.2 | 86.3 | 42.0 | 0.07002 | 0.2449 | 0.02845 | 18.23 | 76.26 | 2300 | 0.6391 | 0.6363 | 0.8030 | 0.8055 | 35.27 | 35.10 | 2275 | 2260 |
| *H*-5-F | 1110 | 29.168 | 14.252 | 0.380 | 0.010 | 14.262 | 83.8 | 70.8 | 88.1 | 39.0 | 0.06981 | 0.2448 | 0.02854 | 18.96 | 79.47 | 2508 | 0.7500 | 0.7493 | 0.9435 | 0.9465 | 41.45 | 41.40 | 2465 | 2463 |
| *H*-6-F | 1195 | 29.168 | 14.252 | 0.420 | 0.011 | 14.263 | 84.5 | 71.3 | 89.4 | 41.0 | 0.06962 | 0.2449 | 0.02860 | 19.70 | 82.24 | 2710 | 0.8670 | 0.8651 | 1.0940 | 1.0950 | 48.10 | 48.00 | 2656 | 2653 |
| *H*-7-F | 1273 | 29.168 | 14.252 | 0.510 | 0.013 | 14.265 | 85.8 | 72.3 | 90.8 | 41.0 | 0.06944 | 0.2453 | 0.02868 | 20.37 | 84.91 | 2910 | 0.9958 | 0.9955 | 1.2465 | 1.2490 | 55.45 | 55.45 | 2855 | 2855 |
| *H*-8-F | 1375 | 29.170 | 14.252 | 0.550 | 0.015 | 14.267 | 86.5 | 73.2 | 92.1 | 40.5 | 0.06927 | 0.2454 | 0.02873 | 21.02 | 88.00 | 3120 | 1.1498 | 1.1481 | 1.4380 | 1.4370 | 64.00 | 63.95 | 3062 | 3060 |
| *H*-9-F | 1456 | 29.168 | 14.252 | 0.630 | 0.017 | 14.269 | 87.5 | 74.0 | 93.4 | 40.0 | 0.06910 | 0.2455 | 0.02880 | 21.71 | 90.48 | 3330 | 1.3028 | 1.3097 | 1.6330 | 1.6300 | 72.70 | 73.00 | 3265 | 3270 |

# APPENDIX NO. 3

## CHARTS SHOWING WEIGHTS PER CUBIC FOOT AND SPECIFIC HEATS OF MIXTURES OF AIR AND WATER VAPOR

80 As explained in the paper, the accurate comparison of results obtained by the Thomas meter and the pitot tube depends upon the use of correct values of the properties of air. It was therefore necessary to make a thorough study of this subject, the results of which are presented in this appendix. The formulæ and values were gathered from various sources and represent the most modern information in regard to mixtures of air and water vapor.

81 As the calculations involved are long and tedious, the author has devised and constructed charts which are here presented in the hope that they may be of value to others. The charts were originally drawn to a much larger scale on a tracing about 2 ft. wide by 5 ft. long.

### OUTLINE OF CALCULATIONS TO ACCOMPANY CHARTS D AND E

$P_t$ = total pressure of mixture $= p_a + xp_w$ in lb. per sq. in.
$p_a$ = pressure of dry air in lb. per sq. in.
$p_w$ = saturated vapor pressure in lb. per sq. in. (Marks and Davis steam tables used).
$x$ = per cent humidity
$t$ = temperature of air in deg. fahr.
$W$ = weight of cu. ft. of a mixture of air and water vapor at $t$ temperature; $P_t$ pressure; and $x$ per cent humidity
$w_a$ = weight of cu. ft. of dry air at a pressure of $(14.0 - xp_w)$ lb. per sq. in.
$w_c$ = correction to be added to $w_a$ for pressures above 14.0 lb. per sq. in.
$w_w$ = weight of water vapor contained in 1 cu. ft. of saturated air
$S$ = specific heat of a mixture of air and water vapor (B.t.u. per lb.)
$S_a$ = specific heat of dry air $= 0.24112 + 0.000009\,t$ (Harvey N. Davis, Trans. Am. Soc. M. E., vol. 30, p. 750, 1908)
$S_w$ = specific heat of water vapor $= 0.4423 + 0.00018\,t$ (Willis H. Carrier, Trans. Am. Soc. M. E., vol. 33, p. 1016, 1911)
$R = 53.35$
$T = 459.6 + t$

For Dry Air:

$$p_a V_a = RT$$

$$w_a = \frac{1}{V_a} = \frac{p_a}{RT}$$

$$W = w_a + xw_w + w_c$$

$$w_a = \frac{(144)\ (14.0 - xp_w)}{(53.35)\ (459.6 + t)}$$

$$w_c = \frac{(144)\ [Pt - 14.0]}{(53.35)\ (459.6 + t)}$$

$$S = \frac{(w_a + w_c)\ (S_a) + (xw_w)\ (S_w)}{w_a + xw_w + w_c}$$

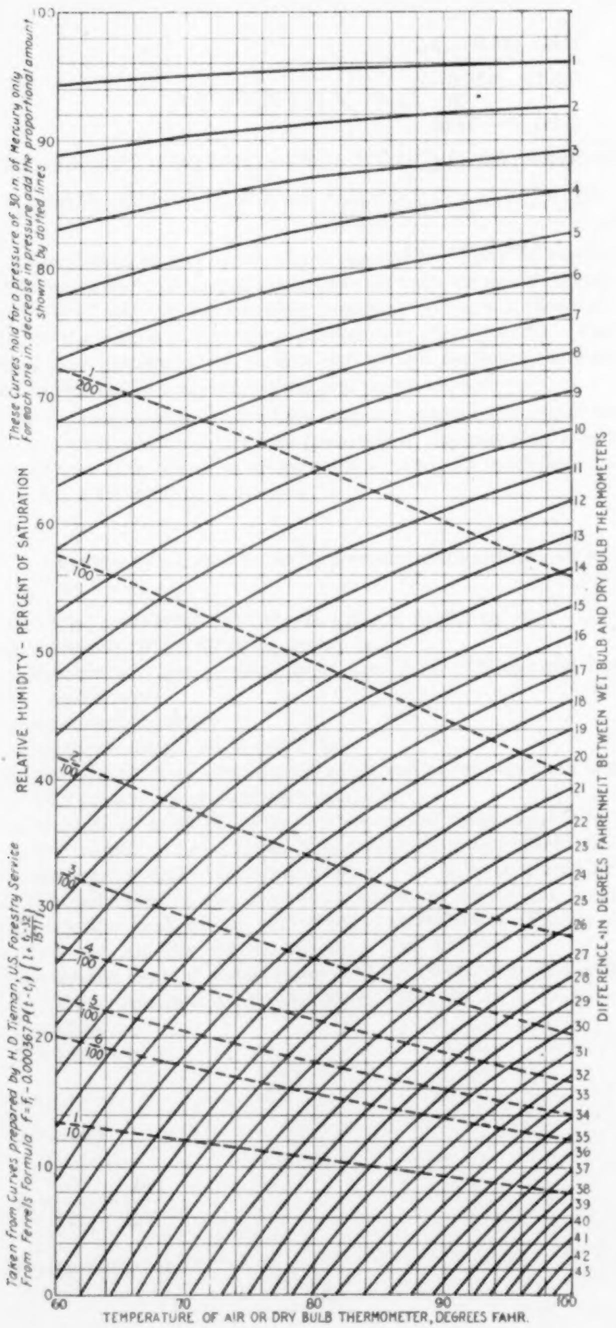



CHART A HUMIDITY CHART

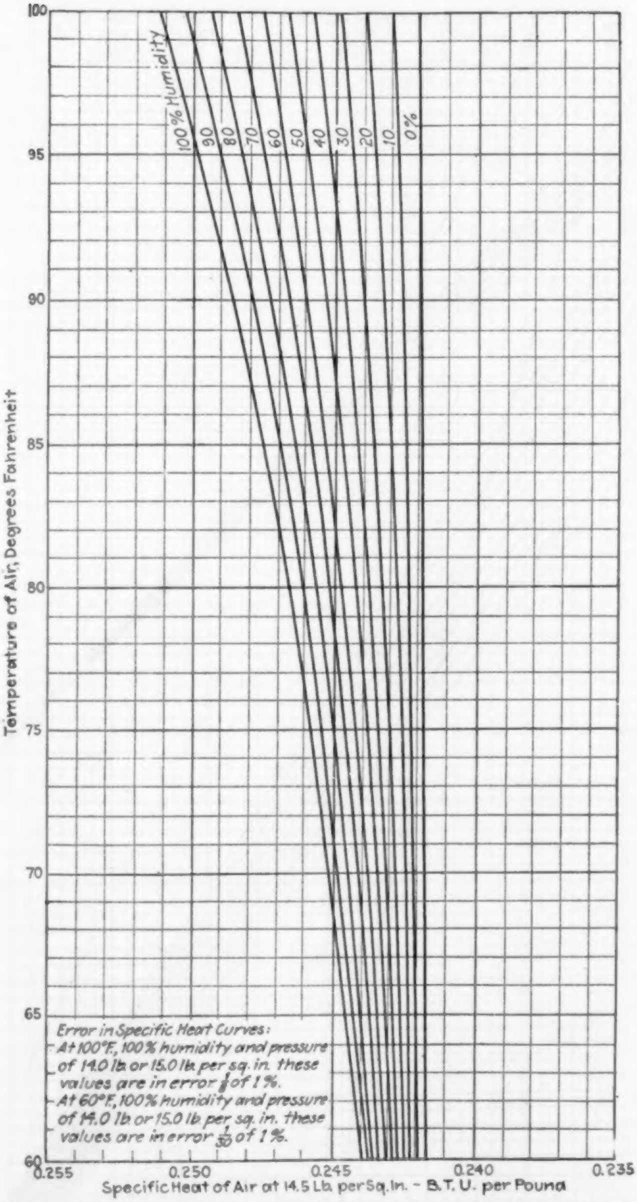



CHART B CHART GIVING THE SPECIFIC HEAT OF MIXTURES OF AIR AND WATER VAPOR AT 14.5 LB. PER SQ. IN. ABSOLUTE PRESSURE FOR VARYING CONDITIONS OF TEMPERATURE AND HUMIDITY

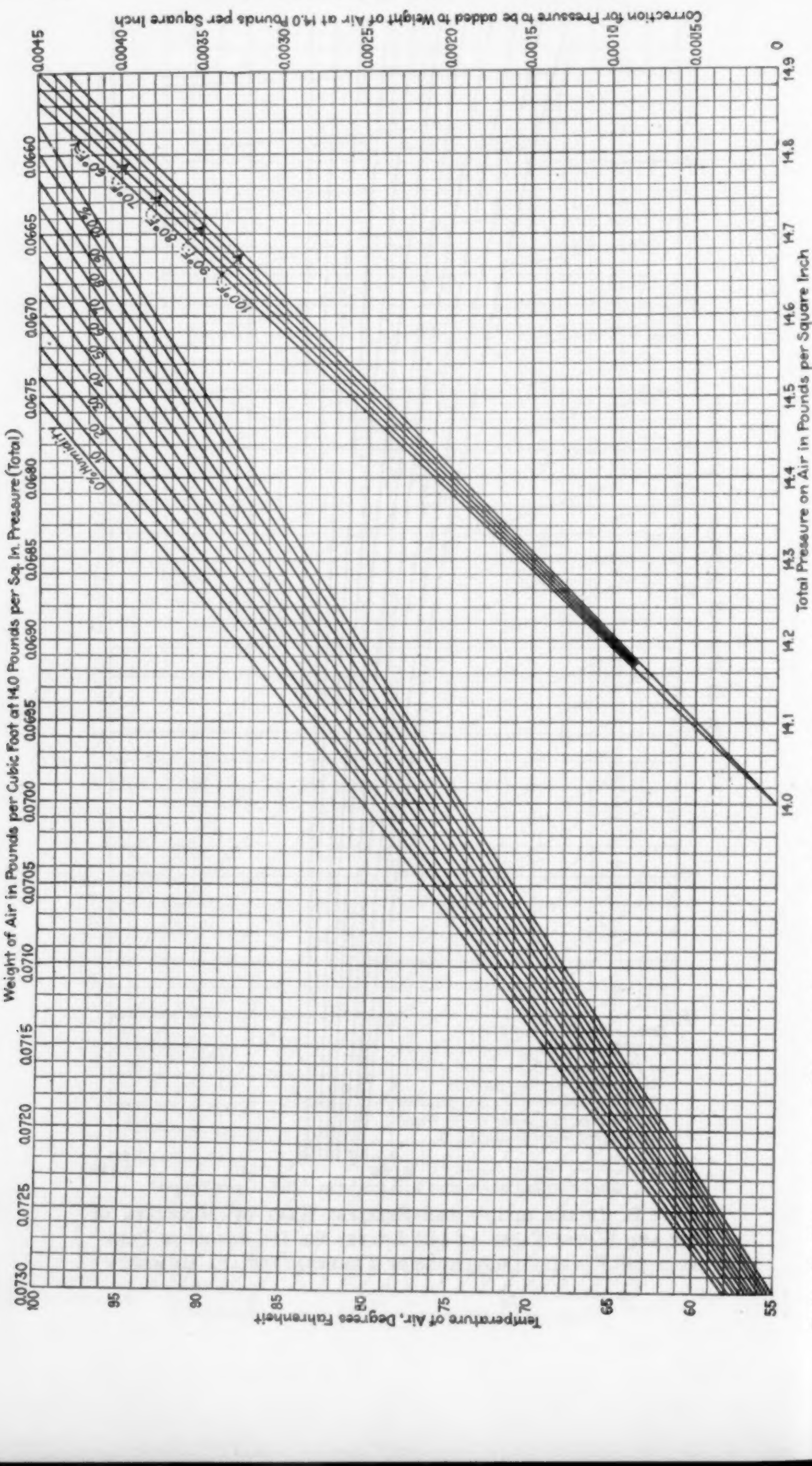



CHART D CHART GIVING THE WEIGHT OF MIXTURES OF AIR AND WATER VAPOR AT 14.0 LB. PER SQ. IN. ABSOLUTE PRESSURE FOR VARYING CONDITIONS OF TEMPERATURE AND HUMIDITY.

CHART E CHART GIVING THE VALUES TO BE ADDED TO THE VALUES TAKEN FROM CHART D FOR ABSOLUTE PRESSURES ABOVE 14.0 LB. PER SQ. IN.

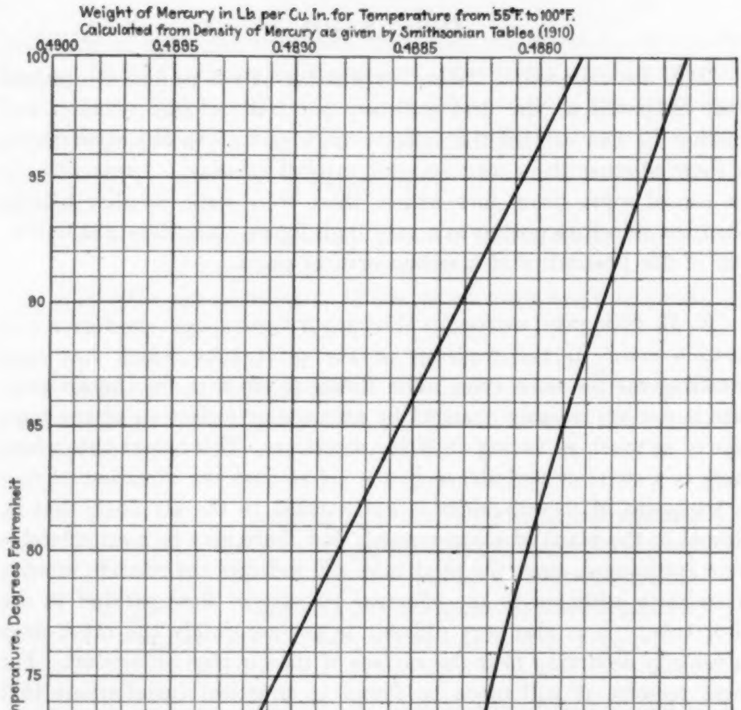



CHART F WEIGHT OF MERCURY IN POUNDS PER CUBIC INCH AT VARYING TEMPERATURES

CHART G WEIGHT OF WATER IN POUNDS PER CUBIC INCH AT VARYING TEMPERATURES

## DISCUSSION

D. S. Jacobus asked the author whether or not he had determined what happened at the first ordinary point of critical velocity, and whether he had carried the velocity high enough in his experiments to show whether there was a second critical velocity. Apparently, in the use of some steam flow meters there were some peculiar actions when the velocities got up to a very high figure, and there was indication of the possibility of a second critical point.

W. H. Carrier (written). The paper throws light on a source of possible error in the readings of the pitot tube which has been troubling me for some time. Mr. Rowse shows that the impact pressure is not appreciably changed by an angular deflection of the pitot tube of as much as 15 deg. in either direction. This means that where there is a swirl or turbulence in the air so that the direction of flow at the point of measurement is not parallel to the air duct, that is, normal to the plane of measurement, that there may be a considerable error introduced, since the pitot tube will indicate the relative velocity of swirl in addition to the effectual velocity of flow parallel to the pitot tube. It is also very difficult to get accurately the rapid drop in velocity occurring near the surface of the air pipe or conduit. For these reasons, it will often be found in practical installations that even with a very careful use of the pitot tube, apparent air quantities will be obtained which are 5 per cent or more higher than the actual.

Of course in a laboratory or shop test, as conducted by Mr. Rowse, where comparatively long straight piping is used and special means are taken to insure substantially parallel flow, as well as uniformity of velocity, the results given by the pitot tube will be exceedingly accurate, as his tests indicate. In practical fan installations, however, it is usually impossible to obtain anywhere near such favorable conditions for accuracy of measurement. In about nine cases out of ten it is necessary to take the measurements either at, or close to, the fan outlet, where there is a great deal of turbulence in flow, as well as a great variation in velocity. In all such cases too great a reliance should not be placed upon the accuracy of the pitot tube.

On account of this inherent source of error in the form of pitot tube advocated by Mr. Rowse, it should be adopted as a standard instrument only until some more reliable and perfect form is devised. Such an improved form of pitot tube should give an impact or differential reading which would vary properly with the angle of

deflection of the pitot tube with respect to the direction of velocity; that is, for the accurate integration of effectual velocity in turbulent flow, the effectual velocity pressure should correspond substantially to the following formula at various angles of velocity direction

$$p_e = p', \cos^2 a$$

where

$P_e$ = the effectual velocity pressure indicated by the differential gage and pitot tube

$P'$ = the actual velocity of the air at some angle to the tube

$a$ = the angle of deflection of the pitot tube with respect to the actual direction of the air velocity

It would seem quite possible that with some experimentation such a reliable form of pitot tube could be produced. This could be accomplished in either one of two possible ways:

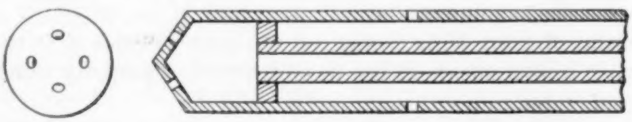


Fig. 27 Special Form for Impact End of Tube

*a* By constructing the impact member so that it will be sensitive to the change of angle in any direction, as shown in Fig. 27.

*b* By constructing the static member so that it will increase the static pressure with any change of angle sufficient to compensate for the error in the impact member

Karl Nibecker asked the author if he had found a critical velocity at which the pitot tube ceased to be accurate. Recently in some work with turbo-blowers where the machine was working at high pressure, the only means of discharge for the air was a comparatively small pipe. The velocity in this pipe, where it was necessary to measure the air, exceeded, by a considerable amount, any velocity that had been attained before within the speaker's experience and which had been measured with the pitot tube. It occurred to him, that very probably there was a point where the velocity was such that the pitot tube could no longer be considered an accurate means of measurement, although it has been found to be accurate for low velocities. In the case in question, the pressure was also much higher than ordinarily obtained

in pipes supplied by simple fans. The pressure in the line to be measured was about 30 lb. per sq. in. and the velocity was in the neighborhood of from 1000 to 1200 ft. per sec.

C. C. THOMAS (written). Referring to *c* under Limits of Accuracy, I understand that this refers to the static pressure only, and that the curves in Fig. 16 indicate that the static pressure is constant over the cross-section of the pipe. If the static pressure is measured by use of the piezometer ring, it appears that the dynamic tube may point at quite an angle with the direction of flow without seriously affecting the reading of velocity head. In this connection, it would seem that *f* under Results, should have the words "due to incorrect static readings" added to the last sentence in the paragraph.

LEO LOEB (written). The paper has served to clear up many of the problems connected with the measurement of gases with pitot tubes, but the conclusions reached in regard to the reliability of certain forms of pitot tubes should not seriously question previous tests involving measurements of the flow of air when properly using these forms of tubes.

In the early comparative tests between the Thomas and venturi meters and the Taylor pitot tube C. C. Thomas reports in a paper[1] that for air velocities between 1500 and 3000 ft. per min., "the pitot, venturi and electric meters will all give accurate and reliable results when properly used under favorable conditions and when observations are taken with a sufficient degree of refinement in method. . . . The close agreement of the results obtained in the measurement of a common quantity by these three fundamental and independent methods of measurements, with independent observations taken for each, forms, from a scientific standpoint, a very interesting proof of the correctness of the several theories involved." The pitot tube used by Professor Thomas is almost the same as Mr. Rowse's type *X*, the slots being respectively 3/64 in. and 1/16 in. wide and in both cases 2 in. long. The general arrangement for test was the same and the fans differed in capacity by the ratio 4 : 3. It appears strange therefore, that while Mr. Rowse calls attention to an average of 9.87 per cent variation between the electric meter and type *X* tube, he does not attempt to reconcile such results with reports from the same laboratory of an average variation of 0.38 per cent between the two meters.

[1]The Measurements of Gases, Journal of the Franklin Institute, November 1911, p. 411.

The Taylor tube with slots 1/16 in. wide and 2½ in. long was used in the measurement of air delivered through a 12½-in. duct to an air heater tested at the United States Naval Engineering Experiment Station in May 1912. Except for the radiation losses, which were determined by a separate test, the apparatus also formed an air meter of the heater type in which steam was the heating medium. While the object of the test was to investigate the heat transfer of the apparatus, the results furnish a sufficient check on the accuracy of the air measurements to be presented in this connection. All tests were of 30 minutes duration with observations every 5 minutes.

| | | | | | |
|---|---|---|---|---|---|
| 1 | Test number | I | II | III | IV |
| 2 | Barometer, in. of mercury | 30.42 | 30.38 | 30.49 | 30.42 |
| 3 | Static air pressure, in. of water | 0.64 | 1.91 | 2.62 | 3.91 |
| 4 | Absolute air pressure, lb. per sq. in. | 15.05 | 15.07 | 15.14 | 15.16 |
| 5 | Room temperature, deg. fahr. | 71.3 | 64.3 | 67.0 | 58.2 |
| 6 | Air entering heater, deg. fahr. | 37.9 | 35.9 | 38.2 | 39.1 |
| 7 | Air leaving heater, deg. fahr. | 144.8 | 130.0 | 128.9 | 123.8 |
| 8 | Relative humidity entering air | 79.9 | 74.0 | 66.1 | 65.1 |
| 9 | Weight of moist air, lb. per cu. ft. | 0.08098 | 0.08148 | 0.08151 | 0.08142 |
| 10 | Mean specific heat of air, B.t.u., per lb. | 0.2477 | 0.2457 | 0.2454 | 0.2452 |
| 11 | Steam pressure, lb. gage | 96.5 | 96.6 | 96.5 | 96.1 |
| 12 | Steam quality, per cent | 0.9879 | 0.9918 | 0.9957 | 1.9 (superheat) |
| 13 | Average air velocity by pitot tube, ft. per min. | 956 | 1677 | 1992 | 2420 |
| 14 | Lb. of air per min. | 66.00 | 116.43 | 138.00 | 168.94 |
| 15 | Steam condensed per min., lb. | 2.050 | 3.092 | 3.517 | 4.058 |
| 16 | Heat yielded by steam, B.t.u., per min. | 1792 | 2710 | 3091 | 3591 |
| 17 | Heat absorbed by air, B.t.u., per min. | 1748 | 2691 | 3072 | 3508 |
| 18 | Heater efficiency | 0.976 | 0.993 | 0.994 | 0.977 |
| 19 | Radiation and losses, B.t.u., per min. computed from radiation test | 7 | 7 | 6 | 9 |
| 20 | Pitot tube coefficient | 0.980 | 0.996 | 0.996 | 0.980 |

The above are only four of a series of 20 tests, all of which verified the results that the tube coefficient varied from 0.98 to 1.00. The average velocity was obtained by a traverse on two diameters using 22 points for observations on each diameter.

A form of inclined manometer which has been used with considerable success is shown in Fig. 28. It consists of a U-tube secured in such a way as to permit the two legs to remain always in the same vertical plane at the same time allowing them to take any desired inclination. The scale is a carefully selected sheet of cross-section

paper. This manometer uses as a fluid a mineral oil of density 0.825 and forms its own leveling device, since there is visible both a distinct vertical and horizontal meniscus. The horizontal meniscus of the two tubes is leveled to the same horizontal line on the cross-section paper by swivelling the backing board, and readings are taken on the horizontal scale. The net reading multiplied by the inclination and the oil density is the true head. The scale multipliers are 0.2, 0.5 and 1.0, which with a 12-in. tube cover the entire range of pressures encountered with forced draft blowers.

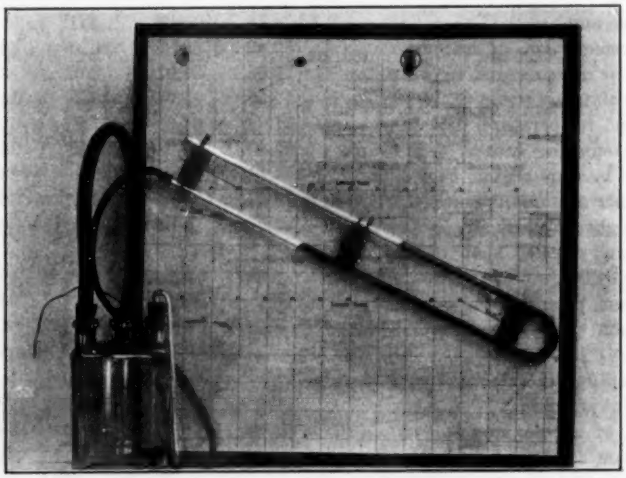


Fig. 28 Inclined Manometer

There seems to be a difference of nearly 2 per cent on the average between results with tubes *H* and *Y* although the dimensioned sketches show almost identical proportions, differing only slightly in overall length and lead connections at shank.

There are cases where the Thomas meter could not be used on account of its introducing resistances against which the fan would not operate in practice. An alternate method of measuring the discharge would be a simple outlet nozzle traversed with an impact tube at sufficient points to give an average velocity. It is therefore to be regretted that Mr. Rowse did not combine with his experiments a traverse of the pipe outlet to determine how completely the static pressure was converted into velocity head.

F. R. STILL (written). The writer, as a designer and builder of fans and blowers and deeply interested in the subject of air measurement for nearly 30 years, has developed a number of instruments for the purpose, but finally settled on the one designated as pitot tube *Y* in Mr. Rowse's paper. This tube has been calibrated in many different ways, such as whirling it on a shaft and correcting for the centrifugal force, by measuring the flow through a tube leading out from a gas holder, and in a pipe in which the flow was simultaneously measured by the Thomas meter. These various comparative tests showed this tube to be less than 1 per cent below the actual velocity. The larger the pipe, the less will be the variation from the actual velocity, as the friction along the walls of the pipe is proportionately less in a large pipe than in a small one.

The particular advantage of the pitot tube is that it is more accurate than any other portable instrument available; in fact, there is no other form of device, either portable or stationary, for measuring accurately large or small volumes at high or low velocities in large or small pipes, which at the same time determines the static pressure or resistance to the flow. The latter is quite as essential to the determination of the work performed as is the determination of the velocity or volume.

The accuracy of the results obtained is largely a measure of the care exercised in the preparation for, and the taking of the readings. The author is correct when he says that the further the readings are taken from the fan the longer will be the spiral or wave of the air as it passes through the pipe. When it is necessary to take readings close to the fan, the cross-section of the pipe should be divided into more and smaller zones so as to get a better average.

The author states further that since the velocity varies as the square root of the velocity head, it was necessary to average the square roots of *each of the 20 readings,* and the *square of this average* represented the mean velocity head. This is unnecessary if the cross-section of the pipe is divided into numerous zones of *equal area;* doing this at first simplifies very much the later computations. For instance, if each zone is of equal area and reasonably small, the average result must be the average velocity of all these areas. But if the pipe is divided into *equal divisions* but of *unequal areas,* then the average of the pressure readings will not give the correct average velocity.

It is indeed very gratifying to note the attention air measurement is receiving of late and the efforts being made to arrive at some stand-

ard method of procedure, as well as the design of some simple, standard, portable instrument for such purpose. The thoroughness of Mr. Rowse's investigations and comparisons should establish definitely the best form of pitot tube for accurate results, and not only settle the ever recurring question as to the accuracy of any type, but also settle once and for all the relative accuracy of the different types, which latter has never heretofore been thoroughly established in the minds of but few engineers.

G. F. Gebhardt (written). The pitot tube with beveled-end static opening originated, the writer believes, in the laboratories of

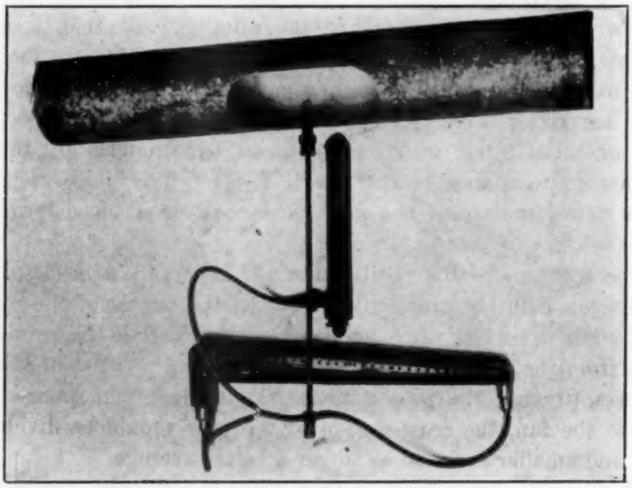


Fig. 29 Tube applied to Circular Conduit

the Armour Institute of Technology. At least no information pertaining to this style of tube was available when the first experiments were made in the year 1904. The sole object of beveling the end of the static tube is to permit of a compact, portable construction which may be readily introduced into a conduit through a single small opening. The application of a commercial design of this tube to a circular conduit is shown in Fig. 29.

Mr. Rowse's conclusion that the beveled-end static opening is not reliable, is true only where extreme accuracy is desired. According to his own experiments the *average* departure of the beveled-end

pitot tube from that of the pitot tube with piezometer static, which he considers reliable, is less than 0.7 per cent for full gate and a trifle above 0.6 per cent for half gate opening (Table 14). The *maximum* departure is 1.6 per cent for full gate and 1.3 per cent for half gate opening. The average departure from the Thomas meter readings is 1.8 per cent for full gate and 1 per cent for half gate opening. Tests conducted by Mr. Kneeland[1] with tubes furnished by the writer gave results agreeing within 1.5 per cent of volumetric

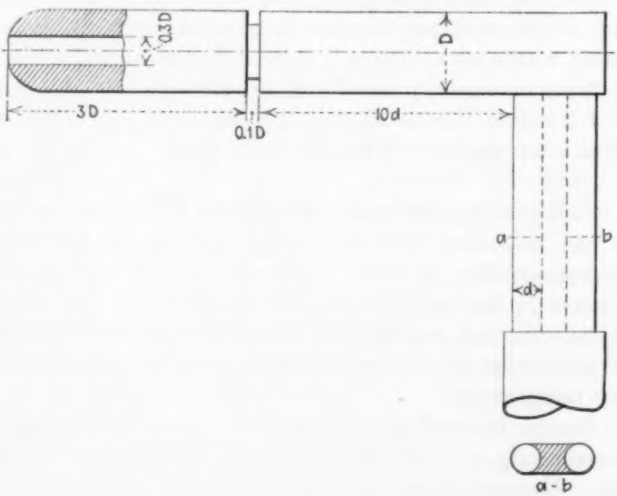



FIG. 30 PRANDTL'S TUBE WITH SPHERICAL END

determinations for all velocities up to 8000 ft. per min. Numerous experiments made in the laboratories of the Armour Institute of Technology on 100 or more tubes with beveled-end static openings gave results agreeing within 2 per cent of holder displacements. For most practical considerations this error is well within the accuracy of other factors involved in determining the flow of gases.

Mr. Rowse is to be congratulated upon the thoroughness of his investigations and for having grouped in a convenient form the relative defects of the various types of pitot tubes commonly employed in measuring the flow of gas in closed conduits.

[1]Trans., Am.Soc.M.E., vol. 33, p. 1155.

C. P. Crissey (written). Engineers interested in the measurement of air and other gases will find much of value in the Rules for Testing Blowers and Compressors approved by the Verein deutscher Ingenieure in 1912, and published in a pamphlet entitled Regeln für Leistungsversuche an Ventilatoren und Kompressoren. Among other things, these rules recommend a tube known as Prandtl's pitot tube. From Fig. 30 it will be seen that this form of tube has a hemispherical head, and a slot extending around the entire periphery. It is stated that a deviation of 15 deg. from the direction of flow has no effect upon the velocity readings, but the static pressure reading is lowered if the deviation is more than a few degrees. Possibly experiments with such a tube will show it has no advantage over some of the tubes so carefully tested and reported upon in this paper, but before a standard tube is adopted it would seem well worth while to investigate the merits of Prandtl's pitot tube.

A. G. Christie (written). Mr. Rowse found that several tubes which had heretofore been regarded as suitable for this work, gave results considerably in error. The care that he took to break up wave action in his test pipe and his comments on this phenomenon would indicate that considerable discretion must be exercised in the use of pitot tubes for the measurement of air or gases discharged by fans or compressors.

In Europe the method generally used for measuring fan or compressor capacity is to discharge through orifices with well rounded mouths. However, there exists a considerable difference of opinion among English and Continental engineers as to the forms of these orifices, their finish and the discharge factors to be applied to them, in spite of the fact that a committee of the Verein deutscher Ingenieure have published a set of rules for power tests on fans and compressors.

In America many different methods are employed in testing fans and compressors, and up to the present time there have been no recognized standard methods or standard test apparatus. Engineers in both Europe and America employ several different expressions for defining efficiency with as many ways of calculating them. Hence, reports of compressor tests are of little use unless the method of calculating them is also available for inspection.

There is an urgent need of a clear, comprehensive statement of means, methods and apparatus to be used in such tests together with an elaborated definition of the expressions of efficiency.

The Author. Dr. Jacobus and Mr. Nibecker have asked questions regarding a critical velocity at which the pitot tube ceases to give accurate readings. No such critical velocity was observed in these tests, the highest velocity reached being about 4500 ft. per min., which was the limit of the apparatus. Tests are now being conducted at the University of Wisconsin with higher velocities and smaller pipes which may provide further information on the subject.

Mr. Carrier's remarks regarding the practical difficulties in measuring fan discharges accurately are very interesting and illustrate the care which must be observed in order to obtain even approximately correct results. Of course, the experiments reported in the paper were conducted under laboratory conditions in order to compare the accuracy of various forms of tubes. The accurate measurement of gases under practical conditions has its own peculiar difficulties, but it is at least something to know that the pitot tube used would give correct results if the conditions were ideal. It is then the engineer's problem to take his readings at such a point and in such a manner as he thinks will give best results.

Mr. Leob has noted that there is a variation of almost 2 per cent between results obtained by tubes $H$ and $Y$, although the tubes are almost identical, which probably represents the possible error of the pitot tube as a means of measuring gases. It would be well to state here that several other tubes of this have been tested at various times at the University of Wisconsin, using this same apparatus, and in every case the agreement between the pitot tubes and the Thomas meter has been so close as to leave in the author's mind no doubt as to the accuracy of this type of tube.

A close study of the tests reported by Prof. C. C. Thomas in the Journal of Franklin Institute, to which Mr. Loeb has referred, will show that the pitot tube used in those tests had small holes in addition to slots; that insufficient precautions were taken to produce parallel flow at the pitot tube; and, most important, that readings were taken on *one radius* only. The readings thus obtained gave the average velocity for only one-quarter of the pipe cross-section, which in this case gave results agreeing closely with the Thomas meter. If readings had been taken across *two diameters* the agreement might not have been so close. Of course it might be possible to make a pitot tube with slots for obtaining the static pressure which would give correct results, although one could not be sure of this fact unless the tube were calibrated.

In answer to Mr. Still's statement that it is unnecessary to average the *square roots* of the observed velocity heads, when the readings are taken at the centers of equal areas, the following mathematical proof is offered:

Let the cross-section of the pipe be divided into five equal annular areas, $A_1$, $A_2$, $A_3$, $A_4$ and $A_5$.

Let the velocity heads at the centers of these annular areas be $h_1$, $h_2$, $h_3$, $h_4$ and $h_5$.

Let the velocities at the centers of these annular areas be $V_1$, $V_2$, $V_3$, $V_4$ and $V_5$.

$$V_1 = K\sqrt{h_1};\ V_2 = K\sqrt{h_2};\ \text{etc.}$$

$$\text{total discharge} = A_1V_1 + A_2V_2 + A_3V_3 + A_4V_4 + A_5V_5$$

or

$$\text{total discharge} = 5A_1\left\{\frac{V_1+V_2+V_3+V_4+V_5}{5}\right\}$$

or

$$\text{total discharge} = (\text{total area})\ (K)$$

$$(\text{total area})\ (K)\left\{\frac{\sqrt{h_1}+\sqrt{h_2}+\sqrt{h_3}+\sqrt{h_4}+\sqrt{h_5}}{5}\right\}$$

A large number of trials from actual tests showed that averaging the velocity heads themselves instead of the square roots gave results from ½ per cent to 1½ per cent too high, the error increasing with the variation between the highest and lowest readings observed.

Professor Gebhardt's remarks concerning the pitot tube having a beveled tube to obtain the static pressure should certainly be taken into consideration in determining the merits of that type of tube.

Professor Thomas has pointed out a result of these tests which the author neglected to emphasize, namely, that for all engineering purposes, the static pressure is constant across the cross-section of the pipe.

Several suggestions have been made as to further experiments with pitot tubes for the measurement of gases which may be summarized as follows:

*a* Tests with low pressures and high velocities. (Such tests are now being conducted at the University of Wisconsin.)

*b* Tests at high pressures and both high and low velocities

*c* Tests to determine the existence of critical velocities

*d* Traverses at pipe opening

*e* Tests using various sizes and shapes of pipes

*f* Experiments to determine a practical substitute for a piezometer, as the piezometer used by the author would be difficult to apply in many cases

*g* Experiments using other forms of the pitot tube, such as the Prandtl tube mentioned by Mr. Crissey

In conclusion the author wishes to second heartily Professor Christie's motion that, just as soon as engineers feel that there are sufficient data at hand, a standard pitot tube for fan tests should be specified and standard procedure and rules for fan tests adopted by the Society.

# No. 1412
# TESTS OF VACUUM CLEANING SYSTEMS

J. R. McColl, Detroit, Mich.

Member of the Society

In June 1911, the firm of engineers of which the writer is a member conducted a series of tests on vacuum cleaning systems for the Board of Education of Detroit, Mich., for the purpose of comparing the merits of the various stationary machines for which proposals had been received, to be used in equipping eight school buildings. The specifications provided that the vacuum cleaning system must be of a capacity such as to take care of two sweepers operating simultaneously on a given system of piping, each sweeper to be provided with 75 ft. of hose. The capacity requirement for each sweeper was stipulated as 80 cu. ft. of free air to be handled with a vacuum of 1 in. of mercury inside an orifice at the tool or sweeper end of the hose. These requirements or specifications for volume, vacuum, and equivalent working orifice, at the end of the prescribed hose, any two of which settle the third, represent what was specified as "the ability to do work."

2 The specifications provided that all bidders must submit their machines for two series of tests: *A*, with the piping system and hose as proposed or required by the manufacturer, for the purpose of determining if under these conditions, the machine would give the required volume and vacuum as above outlined; *B*, with an arbitrary hose and piping system which would be the same for all machines, in order to make a direct comparison of the various machines under identical service conditions. The importance of this second series was emphasized by the specifications. Each of the two series of tests was divided into two parts: (1) measuring the "ability to do work" at the tool end of the prescribed hose: (2) analysis of the machine from the standpoints of maintenance and repairs, simplicity, efficiency of

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

lubrication, dust separation and collection, floor space, noise, tools and hose. Separate grades were given to each machine on its ability to do work and on its character as a machine, and these grades were then combined to get the total score.

3 The specifications provided that in grading machines the ability to do work would not count for more than two-thirds of the total score. It was the sense of the committee who assisted that a machine which makes a fine showing in ability to do work and yet has a low score as a machine is as much to be avoided as one which gains a high score as a machine and shows a poor ability to do work. The two features must be considered together in rating a machine. After going over various details involved, the consensus of opinion was that for a school building, 60 points should be given to the machine as a machine and 40 points to its ability to do work. This was of course an arbitrary standard, regarding which there would no doubt be a wide diversity of opinion among engineers, but as long as all machines were scored uniformly in accordance with this assumed standard, it has the stamp of fairness. The 60 points for the machine as a machine, were divided as follows:

| | | |
|---|---|---|
| Maintenance and repairs | 20 | |
| Simplicity | 5 | |
| Lubrication | 5 | |
| Dust collection | 5 | |
| Space | 5 | |
| Noise | 5 | |
| Tools and hose | 15 | 60 |
| Ability to do work | | 40 |
| | | 100 |

4 Regarding the possible question that may arise in the tests of Series *A* where the machines were not operating under similar conditions as to hose and piping, whether any scores should have been given, on the grounds that it is unfair to grade the results of a machine using a large hose with the results of one working on a small hose, it should be explained that the distinction between the two series of tests is as follows: in the first series, the hose and piping system as proposed was considered an integral part of the machine, inseparable from it; in Series *B* the hose and piping were treated as auxiliary to the machine and independent of it. The scores of Series *A* represent simply the results obtained from the various machines as presented, though working under dissimilar conditions. The scores of Series *B* represent the comparative merits of the machines when

put on the same hose and piping and working under the same conditions.

5 The total grades for the various machines made up from the results of the tests as to the ability to do work, combined with the rating of the machines as machines, are given in column 6, Table 1. The method of computation for that part of the grade covering the ability to do work was made up on the basis of 40 points for meeting full specification requirements of 80 cu. ft. of air per sweeper, with 1-in. vacuum inside the equivalent working orifice. Machines which exceeded this capacity were scored proportionally above 40; and those falling below this capacity were correspondingly debited. It is evident, therefore, that the total score is made up of the sum of the various scores covering the machine as a machine on the basis of 60 points, combined with its ability to do work on the basis of 40 points, when just fulfilling specified requirements. The scores are therefore not percentages, but measures of standing on a reasonable, though arbitrary, scale. The percentages basis could have been used, had no extra credit been given to machines exceeding the specified requirements, or if the capacity attainments of the highest machine were taken as the basis of 40 points and others graded accordingly, but the method used was simpler and equally fair.

6 It was outlined in the specifications that the piping system used as a basis to work from would be a 2-in. line for the inlet system and a 2½-in. for the exhaust system, but manufacturers were permitted to specify other sizes for the first series of tests as a proposed part of their equipment, if so desired or required. This piping system was furnished and installed by the Board of Education. One manufacturer, bidding on the fan type of machine, was permitted to install a 3-in. piping system for use with his machine. Series *B* of the tests included runs made for all high-vacuum machines on the 2-in. piping system, as well as for all machines on a 3-in. piping system, under identical conditions as to hose. It was the writer's expectation to make the comparative series of tests, outlined as Series *B,* with all machines on the largest piping system and the largest hose required or specified by any manufacturer, but he was unable to carry out the plan as regards the hose, and the 1¼-in. hose of the Board of Education was used for all comparative tests.

7 It is obvious that all machines would have given much better results, as regards ability to do work, and lower power consumption, had the comparative tests been made with a larger hose, but in the

TABLE 1 RESULTS OF TESTS

| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | | 10 | 11 | 12 | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Designating Letter | Type of Machine | No. of Sweeper | Size of Piping, In. | Size of Hose, In. | Score | Electric Current Consumption in Kw-Hr. | | Cu. Ft. Free Air for 1 In. Vacuum Inside Orifice | | Vacuum at Machine for 80 Cu. Ft. Free Air per Minute per Hose | Apparent Displacement at Machine for 80 Cu. Ft. Free Air | Pull to Move Following Lengths of Hose on Floor in Lb. | | | |
| | | | | | | For Column 9 | For 80 cf. Fr. Air | | | | | 75 Ft. | 37½ Ft. | 25 Ft. | 12½ Ft. |
| | | As proposed—one-sweeper capacity | | | | | | | | | | | | | |
| $F_1$ | Rotary Exhauster...... | 1 | 2 | 1½ | 93.4 | 2.72 | 3.23 | 86.6 | 2nd test | 7.13 | 106.0 | 18.25 | 8.25 | 5.5 | 3.0 |
| $D_1$ | Multi-Stage Fan....... | 1 | 3 | 1½ | 93.2 | 1.85 | 1.76 | 90.5 | | 3.75 | 92.0 | 22.5 | 9.5 | 7.0 | 3.5 |
| $G_1$ | Single-Stage Fan....... | 1 | 3 | 25′=2¼″+50′=1¾ | 85.6 | 1.62 | ..... | 61.5 | | ...... | ...... | 22.0 | 7.0 | 4.5 | 2.5 |
| $F_1$ | Rotary Exhauster...... | 1 | 2 | 1¼ | 84.0 | 2.15 | ..... | 67.3 | 1st test | ...... | ...... | 13.0 | 5.5 | 4.75 | 2.5 |
| $C_1$ | Rotary Exhauster...... | 1 | 2 | 1¼ | 82.0 | 2.5 | 2.5 | 82.0 | | 7.62 | 108.5 | 19.5 | 9.0 | 4.75 | 2.5 |
| | | As proposed—two-sweeper capacity | | | | | | | | | | | | | |
| $G_2$ | Single-Stage Fan....... | 1 | 3 | 50′=2½″+25′=1¾ | 97.6 | 1.95 | .95 | 177.8 | | 2.07 | 86.0 | 22.0 | 7.0 | 4.5 | 2.5 |
| $F_2$ | Rotary Exhauster...... | 2 | 2 | 1½ | 85.5 | 3.95 | ..... | 142.0 | 2nd test | ...... | ...... | 18.25 | 8.25 | 5.5 | 3 0 |
| $B_2$ | Rotary Exhauster...... | 2 | 2 | 1¼ | 83.0 | 9.25 | 4.22 | 175.5 | 2nd test | 9.87 | 121.0 | | | | |
| $F_2$ | Rotary Exhauster...... | 2 | 2 | 1¼ | 78.7 | 3.99 | ..... | 115.3 | 1st test | ...... | ...... | 13.0 | 5.5 | 4.75 | 2.5 |
| $B_2$ | Rotary Exhauster...... | 2 | 2 | 1 | 66.0 | 10.8 | ..... | 93.8 | 1st test | ...... | ...... | 23.0 | 10.75 | 5.5 | 4.0 |
| $E_2$ | Reciprocating, Plunger | 2 | 2 | 1¼ | 57.7 | 11.5 | 5.37 | 169.0 | | 9.55 | 118.8 | 19.5 | 9.0 | 4.75 | 2.5 |
| $A_2$ | Reciprocating, Plunger | 2 | 2 | 1 | 35.4 | 6.7 | ..... | 97.1 | | ...... | ...... | 25.0 | 13.5 | 8.5 | 5.25 |

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Comparison on school board hose—one sweeper | | | | | | | | | | | |
| $F_1$ | Rotary Exhauster...... | 1 | 3 | 1¼ | 90.3 | 3.11 | 3.1 | 80.8 | 7.3 | 107.0 | | | |
| $F_1$ | Rotary Exhauster...... | 1 | 2 | 1¼ | 88.9 | 3.27 | 3.0 | 78.2 | .... | .... | | | |
| $D_1$ | Multi-Stage Fan....... | 1 | 3 | 1¼ | 86.9 | 1.65 | .... | 59.0 | .... | .... | | | |
| $C_1$ | Rotary Exhauster...... | 1 | 3 | 1¼ | 80.5 | 2.45 | 2.5 | 78.8 | 7.25 | 106.5 | | | |
| $G_1$ | Single-Stage Fan....... | 1 | 3 | 1¼ | 68.7 | 0.7 | .... | 28.0 | .... | .... | | | |
| | | Comparison on school board hose—two sweepers | | | | | | | | | | | |
| $F_2$ | Rotary Exhauster...... | 2 | 3 | 1¼ | 83.9 | 4.67 | .... | 135.0 | .... | .... | | | |
| $B_2$ | Rotary Exhauster...... | 2 | 2 | 1¼ | 83.0 | 9.25 | 4.22 | 175.5 | 9.87 | 121.0 | | | |
| $F_2$ | Rotary Exhauster..... | 2 | 2 | 1¼ | 82.0 | 4.25 | .... | 128.4 | .... | .... | | | |
| $B_2$ | Rotary Exhauster...... | 2 | 3 | 1¼ | 75.3 | 3.5 | .... | 131.0 | .... | .... | | | |
| $G_2$ | Single-Stage Fan....... | 2 | 3 | 1¼ | 72.3 | 1.45 | .... | 69.5 | .... | .... | | | |
| $E_2$ | Reciprocating, Plunger. | 2 | 3 | 1¼ | 61.5 | 12.73 | 6.5 | 183.0 | 10.1 | 122.5 | | | |
| $E_2$ | Reciprocating, Plunger. | 2 | 2 | 1¼ | 57.7 | 11.45 | 5.37 | 168.6 | 9.55 | 118.8 | | | |
| $A_2$ | Reciprocating, Plunger. | 2 | 3 | 1¼ | 48.5 | 5.4 | .... | 148.7 | .... | .... | | | |
| $A_2$ | Reciprocating, Plunger. | 2 | 2 | 1¼ | 45.8 | 5.3 | .... | 139.0 | .... | .... | | | |
| | | 3-in. Piping and proposed hose | | | | | | | | | | | |
| $F_1$ | Rotary Exhauster...... | 1 | 3 | 1½ | 95.3 | 2.58 | 3.15 | 90.8 | 7.1 | 105.6 | | | |
| $F_2$ | Rotary Exhauster...... | 2 | 3 | 1½ | 86.3 | 3.82 | .... | 146.0 | .... | .... | | | |
| $G_1$ | Single-Stage Fan....... | 1 | 3 | 25′ = 2¼″ + 50′ = 1¾ | 85.6 | 0.78 | .... | 61.5 | .... | .... | | | |
| $C_1$ | Rotary Exhauster...... | 1 | 3 | 1¼ | 80.5 | 2.45 | 2.5 | 78.8 | 7.25 | 106.5 | | | |
| $A_2$ | Reciprocating, Plunger. | 2 | 3 | 1 | 34.4 | 6.9 | .... | 94.0 | .... | .... | | | |

writer's opinion, as well as that of one or two of the committee who had much to do with air engineering, the general order of standings would not have been very materially changed, judging from the apparent displacements at the machine, as shown in column 12 of Table 1. The exact relative positions and grades, however, could be determined only by the actual tests.

8 Machines were submitted by seven manufacturers, two submitting both single-sweeper and two-sweeper outfits. A variety of machines was represented; there were two fan-type machines, one a single-stage and the other a multi-stage; two reciprocating plunger-type machines; and three of the rotary exhauster or impeller type. It was interesting to note that machines submitted for the same specified duty, ranged in weight from a few hundred pounds to a ton or more. Another interesting feature was that the vacuum at the machine in the effort to accomplish the same specific requirements as to volume at the tool end of the proposed hose, ranged from little more than 2 in. of mercury to approximately 15 in. The power consumption for the same ability to do work ranged from approximately 1¾ kw. to over 13 kw., depending upon the particular type of machine, and especially on the size of the piping and hose used, which ranged in size from 1 in. in diameter to 2½ in. in diameter, the latter size being that at the inlet connection for a hose tapered down from 1¾ in. in diameter at the tool.

9 No uniformity in sizes of hose and tool handles was found among the manufacturers of so-called high-vacuum machines, and low-vacuum machines, and it will doubtless be a long time before they will agree on an exact size of hose and tool handle. These will always be largely matters of personal taste and also depend on the value put by the manufacturer or user on the question of convenience as contrasted with the importance of the greater ability to do work at a less consumption of power. The larger the hose and tool handle, within reasonable limits, the more rapid and effective work the operator should be able to do with the least consumption of power. But weight of hose and tool handles is often confounded with size. Some of the hose submitted with machines for the tests could have been much lighter and yet have been strong enough for the maximum vacuum on it. The weight of the hose, for convenience' sake, should be as light as possible consistent with durability and the maximum vacuum desired or required to be carried on it, not only for its new

rotund condition, but for its possible flattened condition after months of use.

10 As regards the size of piping, the larger this is, the higher the available vacuum for the hose; if too large, however, the average velocity of the air will not be sufficient to insure the carrying of refuse in horizontal runs, and to prevent clogging. Experience led to the belief that the velocity should not fall much below 2000 ft. per min. in a horizontal run. A tool under ordinary working conditions is handling a varying volume of air. Fortunately at frequent intervals the tool is lifted from the floor or tipped to a marked degree, thereby permitting a comparatively large volume of air to be handled for a brief period at least. If these periods are frequent enough, the volume of air thus intermittently handled will tend to keep the piping system clean, even though the average velocity may be below that already given. Vertical risers should not clog. The factors which operate to limit or prevent the use of large piping are: first cost of installation, the space occupied, the question of cutting into building walls, and the unsightly appearance of large pipes. On the other hand, the enormous wastes due to small hose and piping, so conspicuously brought out in some of the results of the tests, make it desirable to use as large piping as possible, to say nothing of the hose.

11 It seems reasonable to believe that the time will come when the maximum and minimum sizes of hose and piping systems will be standardized for various types of buildings and kinds of service, and that these features will be just as distinctly specified as are now the sizes of wires for a given electric service, or as the sizes of steam pipes and return lines for a given heating service.

## METHODS OF TESTING

12 The Capron school in Detroit, one of the buildings to be equipped with a vacuum cleaning system, was used for the tests and all machines were delivered to the basement for this purpose. The 2-in. piping system in the building and the 3-in. piping system already mentioned were arranged so that each machine in turn could be connected for the various tests, which covered power to drive, ability to do work, vacuum at the machine, speeds, etc.

13 For the work of the committee assisting the author in analyzing the machines as machines, and in grading each according to its merits on the basis of the arbitrary standards given above, each

machine was taken into one of the basement rooms, dismantled and critically inspected from the various standpoints already outlined. Copious notes were made and the various points fully discussed and considered in the subsequent meeting where the final grades were made up.

14 For the power tests at the machine, the usual readings of speed, current consumption, vacuum, etc., were made, similar to those taken in other lines of tests.

15 The apparatus for measuring the ability to do work at the

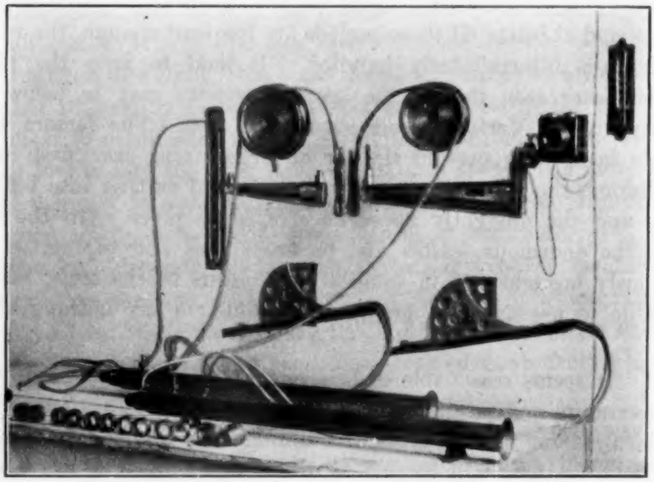


Fig. 1 View of Apparatus for Testing Ability to do Work

tool end of the hose was set up in the principal's office on the first floor and two lines of hose 75 ft. in length were carried from inlet connections on the first floor corridor to this room. As already outlined, the end of each hose was restricted by a series of arbitrary orifices, introduced successively, and the vacua with the corresponding volumes of air handled were determined.

16 The apparatus used in these determinations is illustrated in Fig. 1, taken immediately after the tests. The pitot tube method for determining air velocities was used in the measurement of air volumes. The type of pitot tube used was one which the writer and others who have had much to do in recent years with air measurements have found the most accurate. It was similar to the pitot tube described

in the paper by Mr. Charles H. Treat.[1] Fig. 2 shows a cross-section of the rectifying barrel or tube, pitot and static tubes, orifice, and orifice holder with vacuum gage connection, etc. The rectifying tube consists of a cylinder with an inside diameter of 2 in. and a length of approximately 36 in. The inlet end is made with a bell-mouth to overcome in some degree the vena contracta effect of the entering air, and to assist in making sure that the air is moving practically in parallel lines when it encounters the pitot tube.

17 The pitot and static tubes were soldered together and made into an L-shaped instrument, pointing upstream, and the connections for the manometer were separated for easy connection, as shown. The pitot tube leg was placed parallel to the axis of the rectifying barrel. The dimensions and details of this instrument are shown in Fig. 3. It was first intended to make the pitot tube adjustable to any position across the diameter of the rectifying barrel in order to take a series of readings at properly selected points, and average them as outlined in Mr. Treat's paper. This would give more accurate results as to average velocities, but when the multiplicity of readings for the great number of tests to be made and the limited time available were considered, this idea was abandoned. The average of a series of readings for 80 cu. ft. of air per minute was taken and then the pitot tube was set at the point where it gave the average reading when handling this volume of air. When handling other volumes, the pitot tube reading at this fixed position would not be strictly accurate for determining the average velocity.

18 The static tube consisted of a small brass tube soldered to the pitot tube, the upstream end being sharpened to split the air and cause as little eddying round the instrument as possible. Both pitot and static tubes were made as small as practicable to give as little disturbance as possible in the air currents. But some eddying, of course, is present at best, and to make sure that this should not increase the true static reading, the perforations in the side walls of the static tube were made small capillary holes. Obviously the success of the pitot tube method of air measurement depends, in the first place, on the accuracy with which the pitot and static tubes will separate the pressure which is purely static, from that which is the sum of the static and velocity pressures; and in the second place, on having readings taken at correctly established points which will give average velocity readings for the whole transmitting area.

[1]Trans. Am.Soc.M.E., vol. 34, p. 1019.

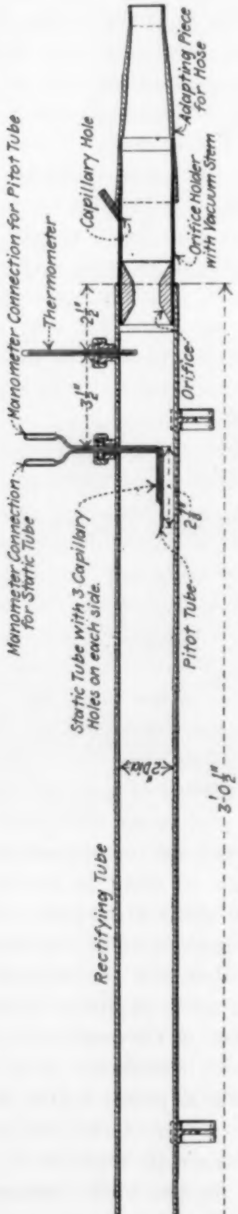



Fig. 2 Section of Pitot Tube Apparatus and Orifice Holder

19 No attempt was made in the design of the orifices used in the tests to make them in accordance with any mathematical formulae. These orifices served only the purpose of restricting the tool end of the hose arbitrarily, in order to give a range of readings between the vacuum behind the orifice and the volume of air handled. If vacuum and orifice or volume and orifice were to be specified instead of volume and vacuum, it would be necessary to give the details of the various orifices to be used, inasmuch as the equivalent working area and apparent area may be at wide variance.

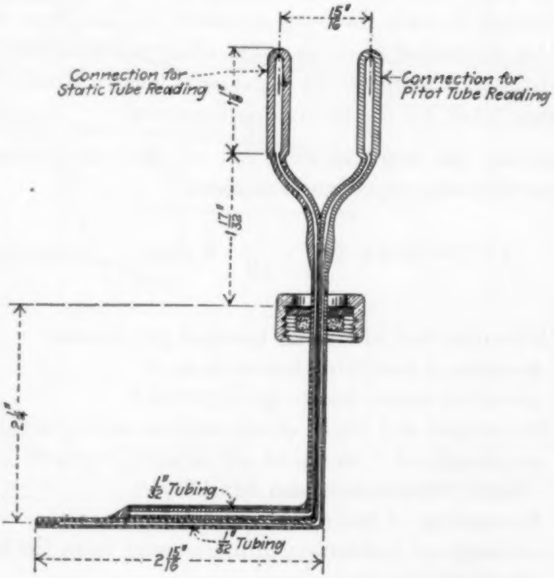



FIG. 3 DETAIL OF PITOT AND STATIC TUBES

20 The inclined manometer with gasolene for liquid was used for determining the difference between total pressure and static pressure or velocity pressure. The manometers are shown in Fig. 1. The angle of inclination for multiplying the reading was taken to suit the velocity pressure developed in order to get as long a reading and as small a percentage of error as possible. Connections to the manometer were reversed and the average taken for each reading so as further to reduce possible errors. The specific gravity of the oil was carefully determined in order that the true value of the velocity head for each reading could be computed. The density of the particu-

lar air handled was computed from government tables, taking into account barometric pressure, temperature and humidity, and from these data, for the average velocity readings, actual velocities were then computed.

21 The average velocity in feet per minute multiplied by the cross-sectional area of the 2-in. rectifying tube, expressed in square feet, gave the volume of air handled per minute for each reading. The rectifying barrel was fitted with a thermometer for taking the temperature of the air passing the pitot tube. Ordinary mercury manometers were used for the readings of the vacua behind the orifices, although in some instances standard vacuum gages were connected up for additional readings. The sling psychrometer was used for humidity and temperature determinations. Weather bureau readings were taken for the barometric pressures.

22 Applying the formula $v = \sqrt{2gh}$ to find the volume of air handled, the following equation is deduced

$$CFM = 60A\sqrt{2g\frac{W}{w} - \frac{1}{12} \quad R\sin\theta} \quad \ldots\ldots\ldots\ldots\ldots \quad [1]$$

where

$CFM$ = cubic feet of free air handled per minute
$A$ = area of rectifying barrel in sq. ft.
$g$ = acceleration due to gravity = 52.2
$W$ = weight of 1 cu. ft. of oil, such as used in manometer
$w$ = weight of 1 cu. ft. of air handled, corrected for barometer, temperature and humidity
$R$ = reading of inclined manometer in in.
$\theta$ = angle of inclination of manometer from the horizontal position

23 For the sake of convenience in reducing the apparent reading of the manometer to the true reading, the manometer supporting plate was graduated so that the manometer could be easily set to positions where its scale reading multiplied the vertical reading by 2, 5, 10, and 20, as desired, according to the particular setting used.

24 In a given test, all the quantities in equation [1] can be taken constant, with the exception of $R$, which alone can be left under the radical, thus simplifying the reduction work for the various readings to a minimum. A scale reading directly in cubic feet of air per minute has been used by the writer as sufficiently accurate for commercial tests.

25 The results of the tests covering the ability to do work, power consumption, etc., are given in Table 1. This table gives the designating letter, type of machine, sweeper capacity of each machine, size of piping and hose used, current consumption and other necessary data including the combined scores made up as outlined. Some of the data given in this table, and other results of the tests, are shown in another form by curves, Figs. 4 to 7. Where the machines did not

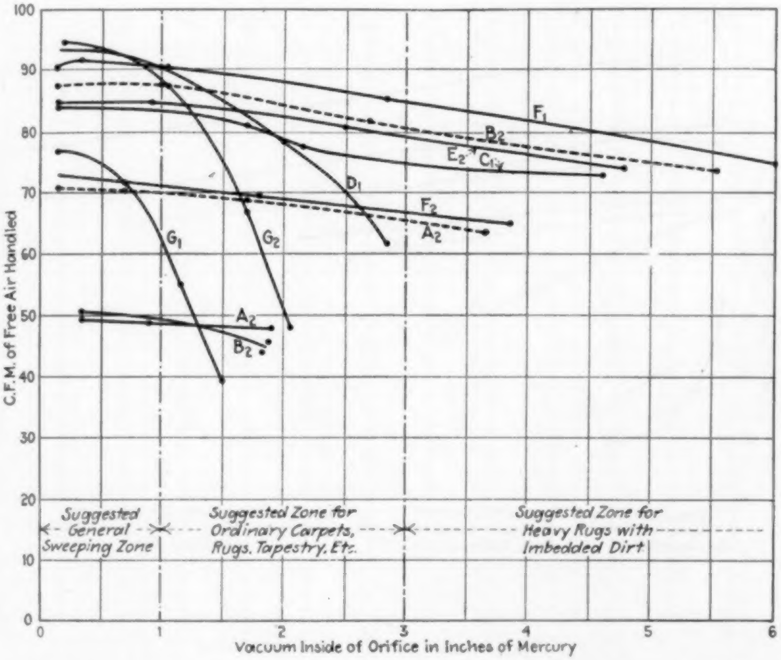



Fig. 4 Curves of Volume and Vacuum at Tool for Piping and Hose as Proposed

handle per hose 80 cu. ft. of free air per minute with a vacuum of 1 in. inside the orifice, the values in columns 8, 10 and 11 are indeterminable.

26 Fig. 4 shows a series of interesting curves, plotted between cubic feet of free air handled per minute and vacuum inside the orifice at the tool end of the hose for the various machines, operating with the piping and hose system as proposed or required by the manufacturer and designated as Series *A*. The types of machines and

sweeper capacities, together with sizes of hose and piping, can be obtained from Table 1.

27 Fig. 5 gives an interesting series of curves plotted between cubic feet of free air handled and vacuum behind the orifice for the various machines operating on the 3-in. piping and 1¼-in. hose; the same for all machines outlined above as Series *B* of the tests.

28 In Figs. 4 and 5 have been drawn zone lines as boundary

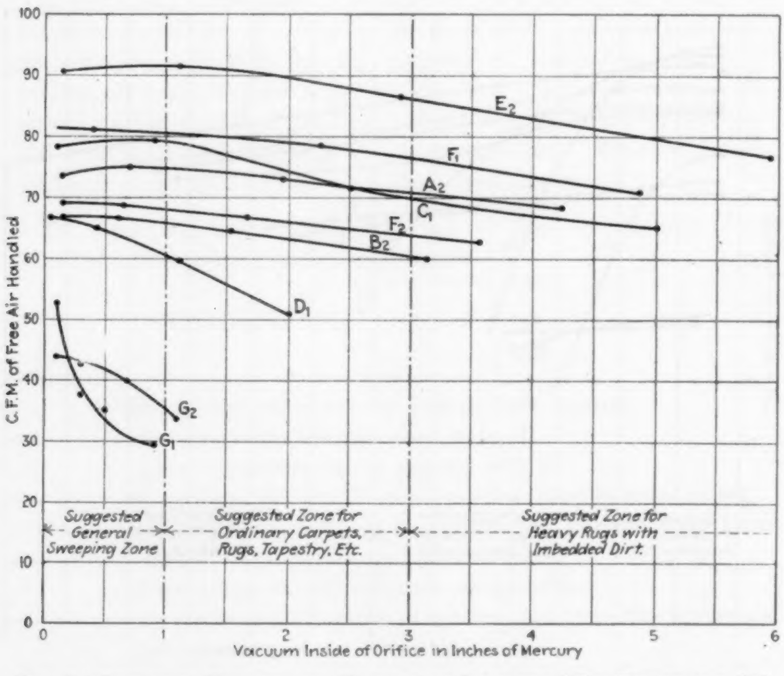



FIG. 5 CURVES OF VOLUME AND VACUUM AT TOOL FOR 3-IN. PIPING AND 1¼-IN. HOSE; THE SAME FOR ALL MACHINES

vacuum lines for effective work in the classes indicated. These are suggestive only, and doubtless would be placed differently by different investigators. It is probable, when vacuum cleaner tests are standardized, that some similar zone boundaries will be established and the qualifications of various machines for different classes of work be thus compared.

29 Without trying to determine whether the expansion of the air through the working orifice or through the piping system is

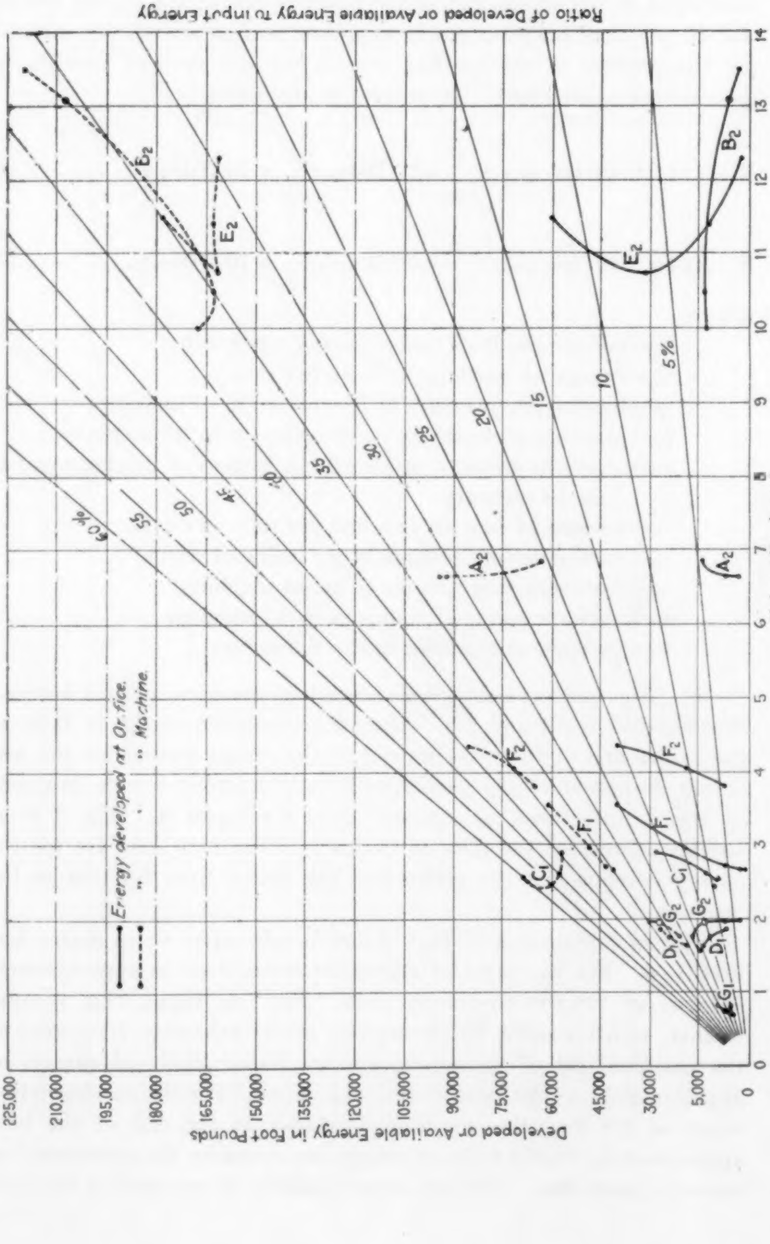



FIG. 6 CURVES, WITH EFFICIENCY LINES, FOR ENERGY DEVELOPED AT MACHINE AND AT TOOL WITH PIPING AND HOSE AS PROPOSED

isothermal, adiabatic or something else, the available energy or work developed at the orifice and at the machine end of the piping system for all practical purposes can be expressed in foot-pounds per minute, by the product of the rarified volume per minute and vacuum in pounds per square foot. To express it algebraically

$$E_1 = 144 \times 0.49\ (p - p_1)\ v\frac{p}{p_1} = 70.704 m_1 v \frac{p}{p_1} = 70.704 m_1 v_1 \ldots\ldots\ldots [2]$$

$$E_2 = 144 \times 0.49\ (p - p_2)\ v\frac{p}{p_2} = 70.704 m_2 v \frac{p}{p_2} = 70.704 m_2 v_2 \ldots\ldots\ldots [3]$$

where

$E_1$ = energy inside of orifice, in ft-lb. per min.
$E_2$ = energy at machine in ft-lb. per min.
$p$ = barometric pressure of free air in in. of mercury
$p_1$ = absolute pressure inside of orifice in in. of mercury
$p_2$ = absolute pressure inside piping system at machine end in in. of mercury
$v$ = volume of free air handled per min. in cu. ft.
$v_1$ = corresponding volume of air inside of orifice
$v_2$ = corresponding volume of air at machine
$m_1$ = vacuum inside of orifice in in. of mercury
$m_2$ = vacuum at machine in in. of mercury

30 Fig. 6 shows curves for the various machines plotted between the electrical input and the developed or available energy in ft-lb. at the orifice and at the machine end of the piping system for the machines connected to the particular hose and piping system proposed by the manufacturer, as outlined above for Series *A*. Fig. 7 gives similar curves for the tests of Series *B*. The full lines are for the energy developed at the orifice and the dotted lines for that at the machine.

31 In explanation of Figs. 6 and 7, take curve $C_1$ in dotted line in Fig. 6. For the point of maximum input there is approximately 2.9 kw. or 128,800 equivalent ft-lb. For this input, this vacuum cleaner, which was the single-sweeper, rotary-exhauster type, gave at the machine end of the piping system, 56,550 ft-lb. of energy, or approximately 44 per cent of the total input. For the corresponding input of 2.9 kw., this machine developed at the end of the hose approximately 28,080 ft-lb. of energy, as shown by the corresponding curve in heavy line. This was approximately 22 per cent of the total

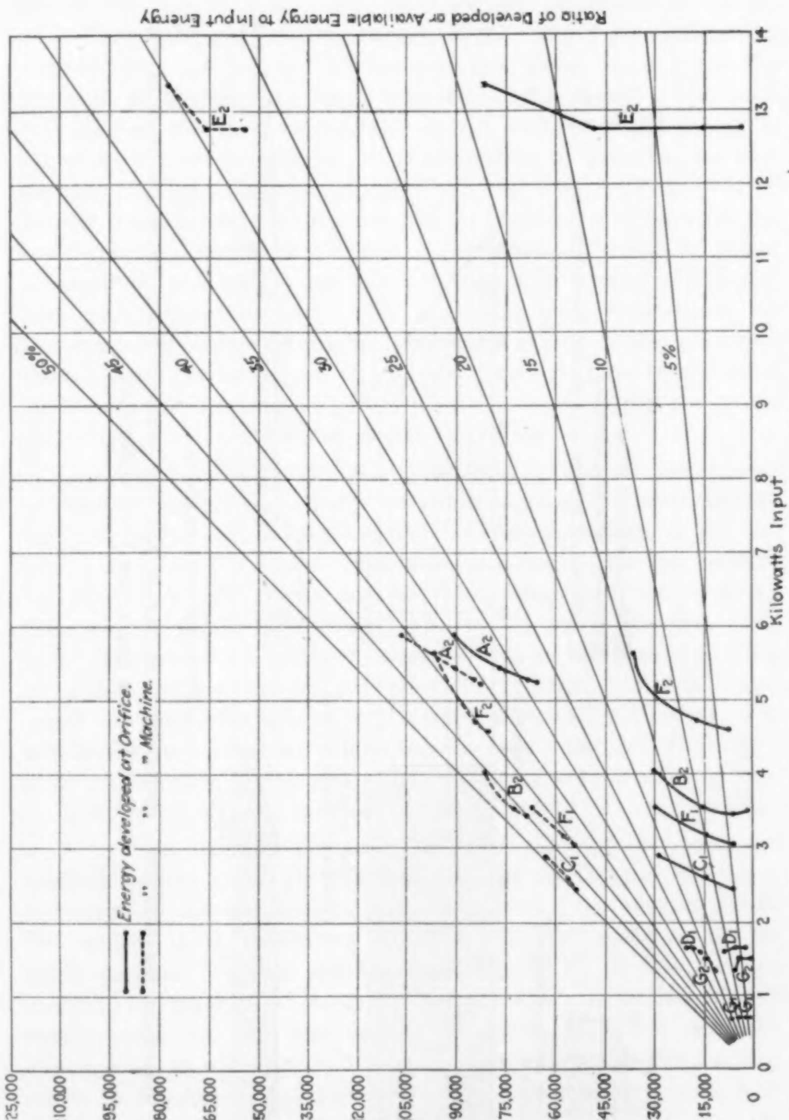



FIG. 7 CURVES, WITH EFFICIENCY LINES, FOR ENERGY DEVELOPED AT MACHINE AND AT TOOL WITH 3-IN. PIPING AND 1¼-IN. HOSE; THE SAME FOR ALL MACHINES

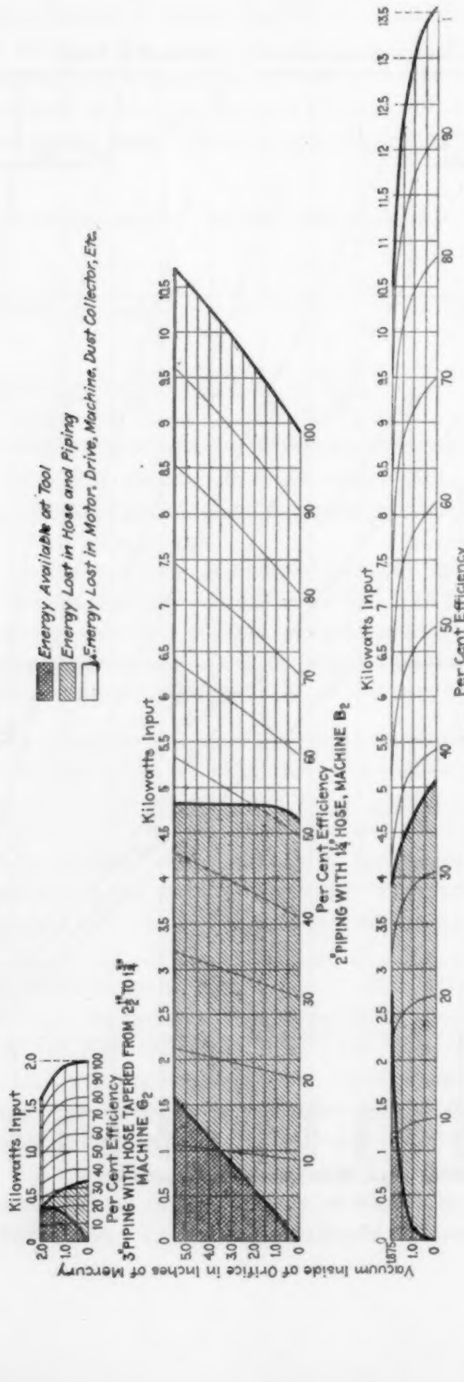



FIG. 8 GRAPHICAL REPRESENTATION OF ENERGY AVAILABLE AT TOOL AND LOSSES IN HOSE, PIPING AND MACHINE FOR A HIGH-VACUUM AND A LOW-VACUUM MACHINE

input, or half the energy available at the machine end of the piping system and represents the condition of greatest restriction of the end of the hose. By similar comparisons, Figs. 6 and 7 give interesting data covering the range of efficiency in production of energy at the tool and at the machine for the various outfits tested. The ratio between the developed or available energy and the input energy is shown by the radiating lines. It is interesting to note that for two-sweeper machines the input ranged from less than 2 kw. to over 13 kw. for machines with piping and hose as proposed to accomplish the same specified results at the tool. The great difference in power consumption was mainly due to the difference in sizes of piping and hose, particularly the latter. To make this more conspicuous the two-sweeper machines having the lowest and highest power consumption as shown by the curves $G_2$ and the curves $B_2$ have been chosen. On these curves are marked conspicuously in larger circles the points coinciding with 1-in. vacuum inside the orifice.

32 Fig. 8 gives for the tests of Series *A*, graphical comparisons of these two machines as regards the power to drive, the losses through machine, piping and hose system and the energy available at the tool end of the hose. The upper diagram is for a single-stage fan machine operating in a 3-in. system and two 75-ft. lengths of hose tapering from 2½ in. at the piping inlet down to 1¾ in. at the tool. The lower diagram is for a rotary exhauster type of machine, operating on a 2-in. piping system with two 75-ft. lengths of 1-in. hose, which was the size submitted by the manufacturers for the tests The middle diagram gives the results for the latter machine when operating on the same piping system but with two 75-ft. lengths of 1¼-in. hose, which was the smallest size permitted by the specifications.

33 These diagrams are made for power consumption losses and available energy under the full range of test conditions. Selecting, for example, the specification requirements of 1-in. vacuum inside the orifice, it is interesting to note that the machine of the upper diagram handled approximately 89 cu. ft. of free air per sweeper; the machine of the lower diagram 49 cu. ft. of free air per sweeper with the 1-in. hose and approximately 88 cu. ft. with the 1¼-in. hose, as will be seen by Fig. 4. The vacuum at the machine for $G_2$ to accomplish this work was only 2 in. of mercury while with $B_2$, with the 1-in. hose, it was 14¾ in. of mercury and with the 1¼-in. hose 10.8 in. of mercury. It is interesting to note for the 1-in. vacuum inside of the orifice, that with $G_2$ less than 2 kw. were

required to drive the outfit and of this energy 68.8 per cent was absorbed in the machine and dust collector, 16.2 per cent in the piping system, and 15 per cent was available at the tool. For $B_2$ using the 1-in. hose, over 13-kw. input was required and of this 65¼ per cent was absorbed in the machine, dust collector, etc., 34.51 per cent in the piping and hose system, giving only 1.24 per cent available energy at the tool. The marked increase shown by the middle figure for $B_2$, using the 1¼-in. hose with other conditions the same, is interesting. There is a total input of 9.25 kw. of which 48.27 per cent is consumed by the machine, dust collector, etc., 48.58 per cent by the piping and hose system, and 3.15 per cent is available at the tool. Changing the two 75-ft. lengths of hose from 1 in. to 1¼ in. decreased the input requirements approximately 30 per cent and increased the available energy at the tool 150 per cent. Similar comparisons can be made for other vacua inside the orifice. Corresponding volumes can be obtained from Fig. 4.

34 Fig. 8 pictures some facts that will be startling to those who have not thought about the marked saving in power and the increased ability to do work that result from the use of larger hose and piping. The clear parts of the diagrams in this figure represent energy absorbed by the machine, including motor and dust collector losses, etc. The hatched portions of the diagrams represent losses in hose and piping. The double-hatched parts of the diagrams represent energy available at the tool. The important query is: What would the machine $B_2$ do if it were put on the piping and hose system of $G_2$?

**The tests to determine the ability to do work, including power consumption, were conducted by Prof. H. C. Anderson and the writer. For a careful analysis of the machines as machines, the author is indebted to a committee consisting of Prof. John R. Allen, of the University of Michigan; Mr. Charles H. Treat, chief designer of the American Blower Company, and Mr. Howard E. Coffin, vice-president and chief engineer of the Hudson Motor Car Company. In the acceptance tests which came a little later, he is indebted to Prof. E. J. Fermier, of the Agricultural and Mechanical College of Texas. All of these gentlemen are members of the Society.**

## DISCUSSION

H. M. Grossman[1] (written). The extreme value of Mr. McColl's paper lies in the fine comparison which the data give of the three different types of vacuum cleaning machines: namely, the reciprocating plunger or piston pump type; the rotary exhauster type; and

[1]Consulting Electrical Engineer, 817 Bellevue Ave., Canton, Ohio.

the high-speed centrifugal fan type of either single or multi-stage design. The development of the vacuum cleaning industry has been very rapid during the past three or four years, and as the industry grew this development has brought out the three distinct types in the following order:

*The Reciprocating Plunger or Piston Pump Type.* This machine consisted of an ordinary cylinder with a reciprocating piston working therein, and with the usual intake and exhaust valves. This plunger or piston pump design was admirably adapted to the production of a high vacuum, but the ability to displace air when the various cleaning tools were in operation was very limited. The fact that the piston pump had always been used when it was necessary to produce a vacuum led naturally to the use of the same device in producing the vacuum for the first vacuum cleaners. In the design and construction of the first type of vacuum cleaners the inventors and builders did not recognize the fact that the volume of air in motion was the essential cleaning agent and that the vacuum was only a means for causing this flow of air.

Since the characteristics of this machine were inherently high vacuum with a very low air displacement (or high-vacuum low-volume) it was equipped with a piping and hose system of very small diameter. The cleaning tools were also made of very small size. The result of this small pipe, hose and tool equipment was constant trouble from clogging of the pipe system, together with a general inability of the cleaning tool to pick up anything except the finest dust. These facts, coupled with rapid wear and tear in the elaborate mechanism of the machine itself, led to almost endless trouble, and required the services of a skilled mechanic to keep it in operating condition. The machine also required an elaborate system of separating tanks to separate every particle of dust from the air before drawing it into the cylinder. Even with the greatest precaution it was frequently necessary to rebore the cylinder after less than a year of use. Naturally, the cost of maintenance of this type of machine was very high. The power required to drive was also very high as compared with the later types.

*The Rotary Pump or Rotary Impeller.* This type of machine was brought out with the idea of obtaining greater air displacement per sweeper with less vacuum maintained at the machine; also, with the hope of eliminating the trouble experienced in the first type due to excessive wear of the piston and cylinder. The second type of machine

required to drive the outfit and of this energy 68.8 per cent was absorbed in the machine and dust collector, 16.2 per cent in the piping system, and 15 per cent was available at the tool. For $B_2$ using the 1-in. hose, over 13-kw. input was required and of this 65¼ per cent was absorbed in the machine, dust collector, etc., 34.51 per cent in the piping and hose system, giving only 1.24 per cent available energy at the tool. The marked increase shown by the middle figure for $B_2$, using the 1¼-in. hose with other conditions the same, is interesting. There is a total input of 9.25 kw. of which 48.27 per cent is consumed by the machine, dust collector, etc., 48.58 per cent by the piping and hose system, and 3.15 per cent is available at the tool. Changing the two 75-ft. lengths of hose from 1 in. to 1¼ in. decreased the input requirements approximately 30 per cent and increased the available energy at the tool 150 per cent. Similar comparisons can be made for other vacua inside the orifice. Corresponding volumes can be obtained from Fig. 4.

34 Fig. 8 pictures some facts that will be startling to those who have not thought about the marked saving in power and the increased ability to do work that result from the use of larger hose and piping. The clear parts of the diagrams in this figure represent energy absorbed by the machine, including motor and dust collector losses, etc. The hatched portions of the diagrams represent losses in hose and piping. The double-hatched parts of the diagrams represent energy available at the tool. The important query is: What would the machine $B_2$ do if it were put on the piping and hose system of $G_2$?

**The tests to determine the ability to do work, including power consumption, were conducted by Prof. H. C. Anderson and the writer. For a careful analysis of the machines as machines, the author is indebted to a committee consisting of Prof. John R. Allen, of the University of Michigan; Mr. Charles H. Treat, chief designer of the American Blower Company, and Mr. Howard E. Coffin, vice-president and chief engineer of the Hudson Motor Car Company. In the acceptance tests which came a little later, he is indebted to Prof. E. J. Fermier, of the Agricultural and Mechanical College of Texas. All of these gentlemen are members of the Society.**

## DISCUSSION

H. M. Grossman[1] (written). The extreme value of Mr. McColl's paper lies in the fine comparison which the data give of the three different types of vacuum cleaning machines: namely, the reciprocating plunger or piston pump type; the rotary exhauster type; and

**[1]Consulting Electrical Engineer, 817 Bellevue Ave., Canton, Ohio.**

the high-speed centrifugal fan type of either single or multi-stage design. The development of the vacuum cleaning industry has been very rapid during the past three or four years, and as the industry grew this development has brought out the three distinct types in the following order:

*The Reciprocating Plunger or Piston Pump Type.* This machine consisted of an ordinary cylinder with a reciprocating piston working therein, and with the usual intake and exhaust valves. This plunger or piston pump design was admirably adapted to the production of a high vacuum, but the ability to displace air when the various cleaning tools were in operation was very limited. The fact that the piston pump had always been used when it was necessary to produce a vacuum led naturally to the use of the same device in producing the vacuum for the first vacuum cleaners. In the design and construction of the first type of vacuum cleaners the inventors and builders did not recognize the fact that the volume of air in motion was the essential cleaning agent and that the vacuum was only a means for causing this flow of air.

Since the characteristics of this machine were inherently high vacuum with a very low air displacement (or high-vacuum low-volume) it was equipped with a piping and hose system of very small diameter. The cleaning tools were also made of very small size. The result of this small pipe, hose and tool equipment was constant trouble from clogging of the pipe system, together with a general inability of the cleaning tool to pick up anything except the finest dust. These facts, coupled with rapid wear and tear in the elaborate mechanism of the machine itself, led to almost endless trouble, and required the services of a skilled mechanic to keep it in operating condition. The machine also required an elaborate system of separating tanks to separate every particle of dust from the air before drawing it into the cylinder. Even with the greatest precaution it was frequently necessary to rebore the cylinder after less than a year of use. Naturally, the cost of maintenance of this type of machine was very high. The power required to drive was also very high as compared with the later types.

*The Rotary Pump or Rotary Impeller.* This type of machine was brought out with the idea of obtaining greater air displacement per sweeper with less vacuum maintained at the machine; also, with the hope of eliminating the trouble experienced in the first type due to excessive wear of the piston and cylinder. The second type of machine

did eliminate all reciprocating parts and substituted a true rotary motion in the vacuum producer, which, in itself, was a decided advantage. The vacuum producer consisted of a casing with a series of close fitting blades or impellers revolving therein. In order to maintain the necessary vacuum the clearance between impeller and casing had to be made exceedingly small, which again caused trouble as soon as a small amount of dust or dirt reached the impeller; therefore, this machine required the same elaborate system of dry and wet separating tanks to protect the vacuum producer from excessive wear. This system used pipe, hose and cleaning tools of almost the same size as the first type mentioned, but the power required to drive this machine was much less.

*The High-Speed Centrifugal Fan Type.* The third and final step in the development of the vacuum cleaner is found in the adoption of the high-speed centrifugal fan, which brings us to the present style of machine, so successful commercially. This type of machine is designed on a new principle entirely, namely, high-volume and low-vacuum. In other words, the vacuum cleaning industry has developed until it recognizes that a large volume of air in motion is required to do rapid and effective cleaning; or, that a comparatively large volume of air in motion is the only cleaning agent, and that the vacuum is nothing more than a means to cause this air to flow.

The machine consists of a high-speed centrifugal fan revolving freely in a snail casing and direct connected to a motor or steam turbine. The volumetric capacity of the fan for displacing air is very large as compared with the first and second types of machines, while the vacuum maintained at the machine is comparatively low. Due to the inherent qualities of a centrifugal fan to displace air, the vacuum maintained at the machine is practically constant regardless of the number of sweepers in operation. The piping system, vacuum hose and cleaning tools are all considerably larger than in the previous types, thus permitting a large volume of air to pass, which, in turn, means a high velocity around the tool lip and a consequent thoroughly effective cleaning of the fabric or floor over which the tool is passed. The separation of the dust and dirt is accomplished by the gravity process, which eliminates all trouble breeding cloth bags and water sprays such as were necessary in the first and second types.

The design and construction of the machine itself is so simple and rugged that the only attention required is an occasional oiling of the bearings and the emptying at intervals of the dirt receptacle. Self-

aligning ball bearings running in a constant oil bath constitute the only frictional or rubbing parts. This type of machine has decided advantages, such as simplicity of design, low first cost, low cost of maintenance, low power consumption, great durability, and ability to do fast and thorough cleaning.

There is another very important point in connection with all vacuum cleaner installations of the past which has been overlooked or neglected in most cases. This point may be summed up in just four words: too much vacuum hose. In other words, the practice has been in the past to use a piping system of so few risers and of such limited extent that it was necessary to use from 75 to 100 ft. of hose per sweeper in order to reach all parts of the building. The vacuum hose is the particular part of an installation that receives the greatest wear and tear; at the same time it is the weakest, and the most expensive part of the installation. Therefore, in order to reduce the amount of vacuum hose necessary, the piping system should be more extensive; that is, provided with a greater number of risers and inlets throughout the building than has been the custom in specifications of the past. The engineer, architect, and building manager, when in a position to do so, should always specify a piping system that will require the use of no more than 50 ft. of vacuum hose per sweeper. This sort of piping installation will reduce the cost of maintenance to a minimum and also make the cleaning of a building a far easier task. In fact, this item alone will reduce the cost of janitor service (including time and repairs) several cents per rentable foot. Therefore, let us always insist on "more risers and inlets, with shorter lengths of vacuum hose."

Charles R. Thurman (written). Mr. McColl has evidently given the matter much thought and has brought out some interesting points in connection with a subject which is rather vague in the minds of many writers of specifications. One point upon which there is a wide difference of opinion among manufacturers and engineers is the quantity of air required to do good cleaning. The author has selected 80 cu. ft. per min. with 1 in. of mercury suction at the end of the hose. This is a good figure for heavy duty, but it is unnecessarily high for ordinary work. A good normal quantity for work of the grade found in school houses would be between 60 and 70 cu. ft. per min. and for residence work, 50 cu. ft. per min. is ample. In all cases there should be a surplus vacuum of approximately 1 in. of mercury above what is required to overcome friction losses.

The author refers to the lack of uniformity among manufacturers in regard to the size of the tools and hose used: The wide variations are indeed astonishing to one who understands the important part played in a vacuum cleaner installation by the hose losses. The writer knows one manufacturer who sometimes uses ¾-in. hose and tools on one of his stationary machines, and another manufacturer who uses 1¾-in. hose and tools. The friction losses in the former are nearly 70 times as great as in the latter, for the same flow of air.

It is the policy of the writer to employ the largest hose that will be acceptable to the average user, provided it can be made with sufficient strength to stand the wear, and flexibility enough not to be unwieldy. The result of this policy has been the adoption of two standard sizes: 1½ in. for office buildings, school houses, etc., where the work is done by male janitors; and 1¼ in. for household work, which is usually done by maids.

Mr. McColl has brought out a very important point in regard to the limiting size of horizontal mains. This is a point which seems to have escaped a great many designers of stationary installations. It is interesting to note that the writer, entirely independent of Mr. McColl, arrived at a value for the minimum permissible velocity in horizontal pipes to prevent settling of the heavier particles of dirt, which was just 10 per cent lower than the figures given in the paper. This is all the more remarkable when the difficulty of determining such a quantity, even approximately, is considered.

Too much care cannot be taken in making specifications for apparatus to be used in the field for determining the performance of machines. For this purpose, the apparatus must be simple, cheap and easily reproduced. The object of such field tests is to ascertain whether the installation meets certain specifications as regards what Mr. McColl has styled "ability to do work." The criterion for such test is usually the suction available at the end of the hose, when the desired quantity of air is flowing. The simplest apparatus for this purpose is a tube to be inserted in the end of the hose and equipped at its outer end with a variety of simple thin plate orifices. A pressure hole is provided in the side of this tube at a distance back from the orifice not less than twelve times the diameter of the tube. In practice, the tube is inserted in the end of the hose and the vacuum measured at the pressure hole with each of one or more sizes of orifice called for in the specifications. This method of test presents an easy means of checking up the "ability to do work" and is one that may be

easily standardized and calibrated so that the flow of air may be computed from the size of the orifice and the vacuum measured.

If the pressure hole is too close to the orifice, the result will be misleading. The writer has often observed a vacuum just back of the orifice considerably greater than that at the exhauster, even when there was a long piece of hose between. This phenomenon is due to the velocity conversion in the tube after the pressure hole is passed.

The zones suggested by the author for the different grades of work do not indicate mature thought. In all vacuum cleaner work, due consideration must be given to the workman doing the cleaning, and experience shows that vacua of more than 3 in. of mercury maintained in the tool impose a heavy burden of labor upon the operator, owing to the adhesion of the tool to the surface. In bare floor tools this difficulty is overcome by various schemes for letting in a rush of air, but for carpet cleaning, such a device defeats the end of the vacuum cleaner by admitting a lot of air, which does not pass through the carpet, and serves only to diminish the vacuum and hence the flow through this fabric. A satisfactory cleaner must have ample air displacement to take care of the fluctuations in demand, due to the rapidly changing action of the tool, and for bare floor work, and must be limited in suction to such a value that the operator is not overworked. In the centrifugal fan type of machine this result is obtained automatically, while in the positive displacement type, recourse must be had to by-passes or pop valves.

The curves showing the relation between the wasted energy and available energy at the tool are very clear. The large percentage of energy absorbed by the piping system and long lengths of hose required by stationary machines make very evident some of the advantages possessed by the portable and so-called truck machines now on the market, in the use of which the piping losses are entirely eliminated and hose losses reduced from 50 per cent to 75 per cent by reducing the length required.

The Author. I fear that Mr. Grossman in his evident enthusiasm for one type of machine has overlooked many good features of the other types, which are far from obsolete at the present writing. It was the prophecy of one of the committee who assisted us, that the relative standings of the reciprocating, rotary pump, and fan types of vacuum cleaners will ultimately be similar to those of the steam prime movers of the reciprocating, rotary and turbine types. This doubtless will be a long way off, for vacuum cleaners will never have

the prominence in engineering and commercial fields that steam engines have had.

The mechanical efficiency of the fan type of vacuum cleaner, working, as it must do, on a fairly low ratio of opening, is much lower, as a machine, than some of the other types, but its greater simplicity is a valuable offset. The low power consumption shown by some of these machines, for a given duty, is a result of large hose and piping rather than high mechanical efficiency of the machine.

The small hose and piping systems used in the earlier vacuum cleaner work, and still used by some manufacturers, required high vacua at the machines, a matter more of necessity than choice, to get practical results in cleaning. There is no reason why these same machines cannot be used, or at least be designed to be used, on a large hose and piping system as "low vacuum—high volume" machines with surprising results in air handled and diminished power consumption. Machine $B_2$ (Table 1) for example had a displacement nearly double that which it would have needed on the hose and piping system of $G_2$. The use of the larger hose and piping systems with the fan type of cleaners, particularly the single stage type, is also more a matter of necessity than choice, and some excellent engineering has been done in adapting these machines (capable of only about 2-in. vacuum as a maximum for a practical unit) to rapid sweeping work.

Piping systems are now quite often installed by heating contractors for adaptability to various kinds of vacuum cleaners, with a view that the machine can be purchased and installed later. If the time ever comes when the hose and tools can be purchased from a general supply house by intelligent selection of engineers and clients, according to the maximum size the user will stand for independent of any particular vacuum of cleaning machine, then we shall arrive at an era when the true comparative merits of vacuum cleaning machines, as machines, will be found out as fully as electric motors are now known, independent of the feed lines, or as steam engines independent of steam and exhaust lines.

My experience with thin plate orifices, outlined by Mr. Thurman, has not been entirely satisfactory, probably due to varying vena contracta effects. It seems to be desirable to get away as much as possible from eddies near the orifice even though the pressure tube connection is through a capillary hole smoothly finished with the inside of the orifice holder. The exact design of orifice and holder must be given if vacuum-orifice or volume-orifice requirements are specified.

No. 1413

# TESTS UPON THE TRANSMISSION OF HEAT IN VACUUM EVAPORATORS

By E. W. Kerr, Baton Rouge, La.

Member of the Society

The object of the tests described in this paper was to secure experimental data regarding the effect of the different factors which influence the transmission of heat in vacuum evaporators.

2 After making a number of tests on multiple evaporators in sugar factories it was concluded that a special experimental plant would have to be constructed for testing purposes, in order to control the conditions of operation. Accordingly, an apparatus was designed and built and, although considerably smaller than the average commercial machine, was large enough to give results comparable with what might be obtained from units of commercial size. This apparatus was erected and operated in the mechanical laboratory of the Louisiana State University and Audubon Sugar School.

3 The transmission of heat through the tubes of evaporators is affected by the following factors:

- *a* Velocity of juice circulation
- *b* Distribution and velocity of the heating steam
- *c* Presence of incondensable gases in the steam compartment
- *d* Hydrostatic head of the boiling liquid upon the heating surface
- *e* Presence of condensed steam upon the heating surface
- *f* Density of the liquid being concentrated
- *g* Cleanliness of the heating tubes.

4 *Velocity of Juice Circulation.* The circulation of liquid in most evaporators is not positive, but depends upon convection currents much the same as in steam boilers, steam tube evaporators being

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

analogous to fire tube, and liquid tube evaporators to water tube boilers in this respect. In view of this fact it is evident that the advantages of high velocity of circulation are not obtained with evaporators to the extent possible with surface condensers, where the circulation of the cooling water is positively controlled by a pump or other means.

5 With submerged tube evaporators, the velocity of circulation can, however, be considerably increased by careful attention to details such as proportions of tubes and the use of circulation tubes, or downtakes. Fig. 4, one of the experimental evaporators used in the experiments, shows such a circulation tube at the center. The boiling liquid ascends through the small tubes and descends through the downtake tube, due to the fact that there is a greater amount of heat transmission per unit of carrying capacity through the small tubes than in the larger circulation tube.

6 *Distribution and Velocity of the Heating Steam.* Of the devices used in commercial evaporators for effecting even steam distribution to the heating surface may be mentioned the annular steam belt surrounding the tubular cluster, with slots through which steam passes radially inward. This has been used very generally with vertical submerged tube evaporators.

7 The vertical submerged tube (standard) evaporator with steam supplied through an annular belt offers little opportunity for securing high steam velocity; in fact, the average steam velocity is very low in this type. One of the evaporators experimented with, shown in Fig. 7, is opposite in principle to the belt type. It was designed to secure high velocity at the expense, however, of considerable friction, instead of low velocity and little friction as in the belt type. The steam tube evaporator gives opportunity for controlling steam velocities within certain limits by proportioning the tubes, a long tube of small diameter giving a greater velocity than a shorter tube of larger diameter.

8 *Presence of Incondensable Gases in the Steam Compartment.* The presence of air or other incondensable gases not only reduces the coefficient of heat transmission because of their resistance to heat conduction, but according to Dalton's law of mixed vapors, produces a temperature in the steam compartment lower than that corresponding to the vacuum shown by the gage and so decreases the temperature fall.

9 Most evaporators are provided with means for removing the

incondensable gases, though in many of them this is done more or less ineffectively because under the agitated conditions incident to the rapid inrush of steam there can be but little segregation. The removal of air must therefore be at the expense of much steam along with it. Incondensable gases are likely to collect in dead spaces which makes the matter of steam distribution of added importance.

10 Fig. 7 illustrates a special device applied to vertical liquid tube evaporators, one object of which is to effect a more efficient removal of the incondensable gases. As the steam passes toward the center of the steam compartment it condenses and gradually becomes richer in air, thus affording a means of separating air from steam, the former being drawn off from the center.

11 Fig. 6 is a steam tube evaporator especially designed to remove incondensable gases effectively. In this case the gases are removed from each steam tube, which is closed at the top, by means of a $\frac{1}{8}$-in. pipe. One type of film steam tube evaporator with horizontal tubes vents the gases directly to the vapor space of the boiling liquid through very small holes in the closed end of the tubes.

12 *Hydrostatic Head of the Liquid upon the Heating Surface.* Hydrostatic head of juice, that is, deep submergence of the heating surface, reduces heat transmission and capacity by decreasing the temperature fall for a given vapor pressure difference. The added pressure due to hydrostatic head increases the average temperature of the boiling liquid, and this in turn decreases the temperature fall. For example, assume the steam pressure in the calandria to be 20 lb. absolute and the pressure of the vapor above the boiling liquid 12 lb. absolute, corresponding to temperatures of 228 deg. and 202 deg. respectively. If the evaporator were of the film type, the actual temperature fall would be 26 deg. If it were of the submerged tube type like that of Fig. 5, with the juice level say 48 in. above the lower tube plate then the total pressure of boiling would be equal to the vapor pressure plus the average static liquid pressure. The latter is equal to $\frac{24}{12} \times 0.43 = 0.86$ and the total pressure to $12 + 0.86 = 12.86$ lb. per sq. in., corresponding to a temperature of 205 deg. in which case the temperature fall would be 228 deg. — 205 deg. = 23 deg. instead of 26 deg., as in a film evaporator. A similar calculation will show that the lower the absolute pressure of boiling, the greater

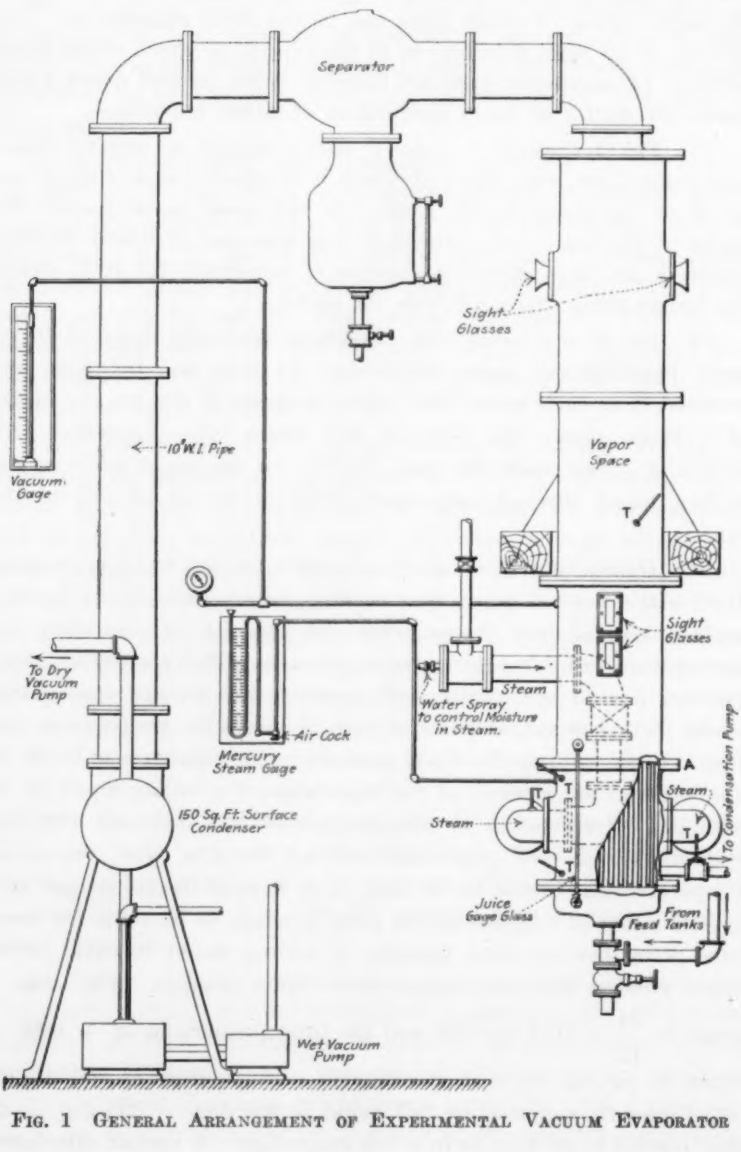



FIG. 1 GENERAL ARRANGEMENT OF EXPERIMENTAL VACUUM EVAPORATOR

the loss in temperature fall with a given depth of submergence, and this means that the loss will be greatest in the last body of a multiple evaporator.

13 *Presence of Condensed Steam upon the Heating Surface.* As with all steam heating apparatus it is important that all condensed steam be removed as it is formed. If allowed to collect in the heating

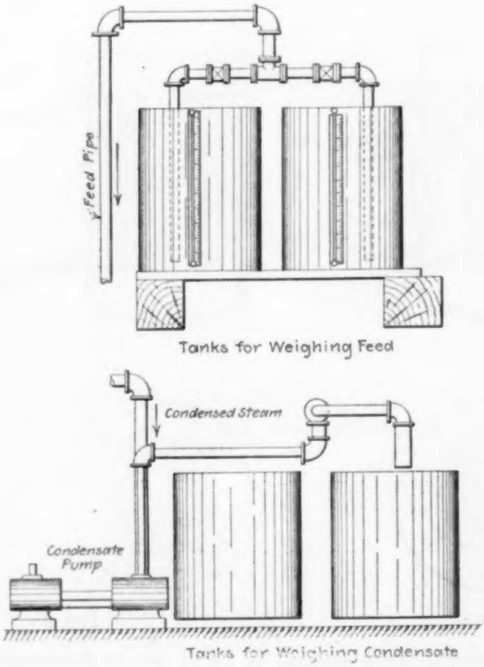



Fig. 2 Weighing Tanks for Experimental Apparatus

compartment the heating surface thus submerged is made less effective since the transmission from liquid to liquid is less than from steam to liquid. In evaporators with long vertical tubes the condensed steam must run down the entire length of the tubes, and this doubtless reduces the heat transmission. With horizontal steam tubes, the water is swept out of the tubes by the current of steam.

14 *Density of the Liquid being Concentrated.* The density of the liquid being concentrated affects heat transmission in that the temperature of boiling is increased above that corresponding to the pressure and this results in decreased temperature fall. The co-

efficient of heat transmission may also be decreased in some cases, though this is a matter of some doubt. The effect of density in this respect is naturally greatest in the later bodies where the absolute pressure is least and where the density is greatest.

15 *Cleanliness of the Heating Tubes.* The effect of the fouling of heating tubes upon heat transmission is too well known to be re-

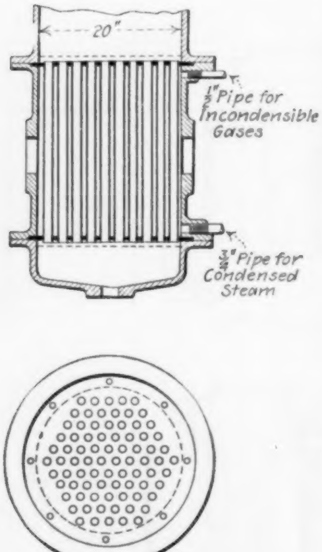



FIG. 3

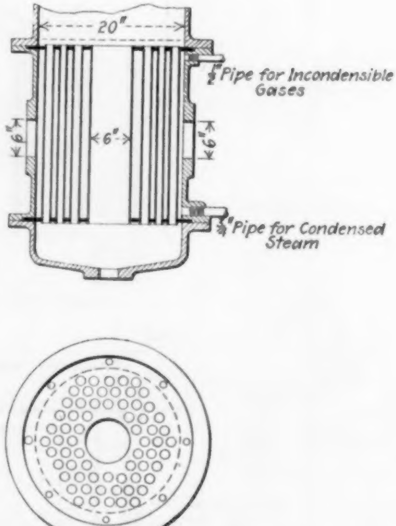



FIG. 4

FIG. 3 80 COPPER TUBES 1⅜ IN. BY 24 IN. BY 18 BIRMINGHAM WIRE GAGE
FIG. 4 66 COPPER TUBES 1⅜ IN. BY 24 IN. BY 18 BIRMINGHAM WIRE GAGE. 6 IN. DOWNTAKE. TUBE PLATES $\frac{3}{16}$ IN. COPPER

marked. The collection of solid matter upon the heating surface depends mainly upon the nature of the liquid and the velocity of circulation. With sugar juices it is generally necessary to stop the evaporation every one or two weeks to clean the tubes. This may be done by boiling out with caustic soda followed by hydrochloric acid. If exhaust steam is used for heating, there is likely to be some fouling due to the lubricating oil contained, especially if cheap oils, which saponify under the action of heat, are used.

16 As it is very difficult to control conditions in the different

bodies of a multiple evaporator, it was decided to use an evaporator with a single body so constructed that the conditions existing in any of the bodies of a multiple evaporator could be reproduced. With this in view the apparatus was so designed that it was possible to change the type of heating compartment and to control at will the pressure of the heating steam, the vacuum under which boiling takes place,

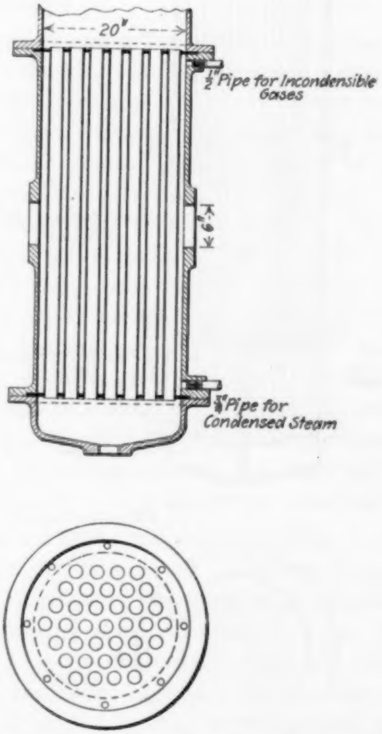



Fig. 5 37 Copper Tubes 2 In. by 48 In. by 18 Birmingham Wire Gage. Tube Plates $\frac{3}{16}$ In. Copper

the quality of the steam, the amount of air in the heating steam, and the density of the liquid.

17 Figs. 1 and 2 show the general arrangement of the plant used in all of the tests. The surface condenser contains 150 sq. ft. of cooling surface, whereas the heating surface in the evaporator itself varied from 50 sq. ft. in the smallest calandria tested up to 86 sq. ft. in the largest. It will be noted that the body of the evaporator is

supported above the calandria so that the calandria can be removed and another one bolted on conveniently. Fig. 1 shows the calandria

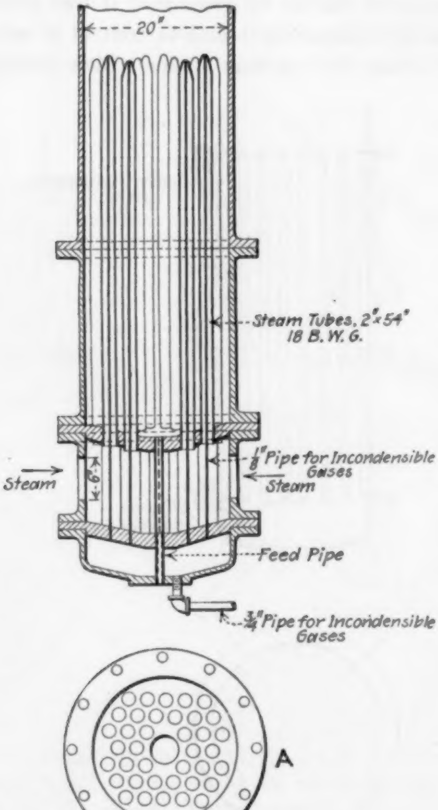



FIG. 6 38 COPPER HEATING TUBES 2 IN. BY 54 IN. BY 18 BIRMINGHAM WIRE GAGE. TOTAL HEATING SURFACE 86.21 SQ. FT.

of Fig. 3 bolted to the body at *A*. In changing from one calandria to another this joint had to be broken.

18 Figs. 3 to 7 inclusive show the different types of calandria tested. The calandria shown in Fig. 4 is like that of Fig. 3 except that there is a 6-in. circulation tube or downtake at the center. Fig. 5 shows another calandria with tubes 2 in. in diameter and 48 in. long without a downtake.

19 Fig. 6 shows an evaporator of the steam tube type with special arrangements for removing the incondensable gases. There are two thick tube plates, one above the other, both of cast iron. The heating tubes which are 2 in. in diameter and 54 in. long are expanded into the upper tube plate. These tubes are closed at the top and are open at the bottom. Into the bottom tube plate ⅛-in. tubes are screwed. These small tubes are open at both ends and are placed inside of the heating tubes, reaching nearly to the top of the latter. The space below the lower tube plate is connected to the vapor space of the succeeding body in an evaporator of commercial size. Steam passes up into the heating tubes in the annular spaces surrounding the gas tubes, the incondensable gases being driven towards the top from whence they are removed by the small gas tubes, each heating tube having its own individual incondensable gas remover. The lower tube plate is made saucer shape so that the condensed steam drains to the center from where it is removed in the usual manner.

20 Fig. 7 shows a calandria similar to Figs. 3 and 4 with tubes 1¾ in. in diameter and 24 in. long. It differs from them, however, in the manner of distributing the steam to the heating tubes, and in the manner of removing the incondensable gases, the steam being supplied through four openings, one above the other, thus giving better distribution vertically. Between the two tube plates a vertical baffle plate is placed so as to guide the steam along a circuitous path to the 3-in. downtake at the center. This baffle plate is so placed that the passage for the steam is gradually reduced in cross-section in order to keep the steam velocity as high as possible, overcoming to an extent, the tendency to decrease the velocity due to condensation. The incondensable gases are drawn off by means of a small perforated pipe through the top tube plate and reaching nearly to the bottom tube plate. The object of this design is to obtain high steam velocity among the heating tubes and effective separation of incondensable gases from steam.

21 The tanks for weighing the condensed steam, or "condensate,"

were used in only a few of the tests in which it was attempted to determine the radiation loss from the surface of the evaporator. The steam used in the evaporator was taken from the power house boiler at a pressure of about 80 lb. gage, the desired pressure in the tube belt being obtained by throttling at the valve.

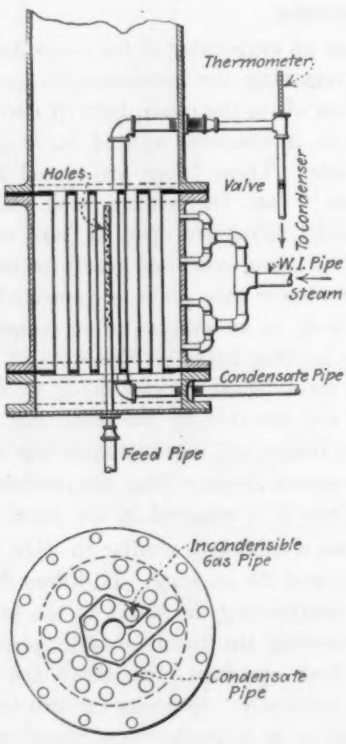



FIG. 7 30 COPPER HEATING TUBES 1¾ IN. BY 24 IN. BY 18 BIRMINGHAM WIRE GAGE. TOTAL HEATING SURFACE 29 SQ. FT.

22 The steam pipe in all of the calandrias except that of Fig. 7 was 6 in. in diameter at the entrance to the tube section and made to enter at two points. In this way the steam was well distributed. The steam being expanded in some cases from 80 lb. down to less than atmospheric pressure, there was considerable superheat. In order to get steam of varying quality, provision was made for spraying water into the steam between the throttling valve and the calandria.

23 In commercial evaporators it is customary to remove the concentrated liquor from the last body by means of a pump. In the experimental outfit a slightly different arrangement was used. In starting a test, sugar juice of the desired density and quantity was placed in the evaporator and as the evaporation progressed water was supplied in such quantities as would keep a constant level in the juice compartment, as shown by the juice gage glass. In this manner conditions sufficiently near those in actual evaporators were obtained with considerably less complexity of apparatus. After being weighed in the calibrated barrels, the water was fed by gravity aided by the vacuum in which the boiling was usually carried on, a valve in the feed pipe being used to regulate it.

24 That part of the evaporator above the top tube plate, which will be designated as vapor space, was 10 ft. high in all of the calandrias tested except that of Fig. 6 in which it was about 8 ft. This liberal height was provided in order to prevent, as far as possible, the carrying over of liquid in the vapors leaving the boiling surface. The separator shown was originally designed to be used as an oil separator, but was here used to catch any liquid entering the vapor pipe.

25 Thermometers were placed as shown in Fig. 1 to measure the temperatures in the steam pipe, the vapor space, the bottom of the steam compartment, the top of the steam compartment, the entering juice or water, the condensed steam and the room. This applies in its entirety to the apparatus when used with the tube sections shown in Figs. 3, 4 and 5. The thermometers were placed somewhat differently for the other tube sections. The mercury manometer connected to the steam compartment could be used with pressure either above or below the atmosphere and was arranged so that the water could be kept out of the mercury column when operating with pressure. With low vacua the wet vacuum pump was sufficient, but with higher vacua, that is, 24 in. or more, it was necessary to use the dry vacuum pump. A majority of the tests was made with water as the liquid to be boiled, the balance being made with sugar solutions produced by mixing white sugar with water in such proportions as would give desired densities. Liquor made in this manner was clean and pure and gave practically no fouling of the heating tubes. All tests were made with practically clean heating tubes.

METHOD OF MAKING A TEST

26 In starting a test the wet vacuum pump was first started and a charge of juice drawn in. The condensate pump was then started and steam turned on to warm the apparatus. After the desired conditions as regards steam pressure, vacuum, height of boiling, etc., were obtained the apparatus was operated for some time before the test was started. The duration of the tests varied from 20 to 60 minutes, depending upon conditions. Each test started with the juice level indicated by a string around the juice gage glass and the

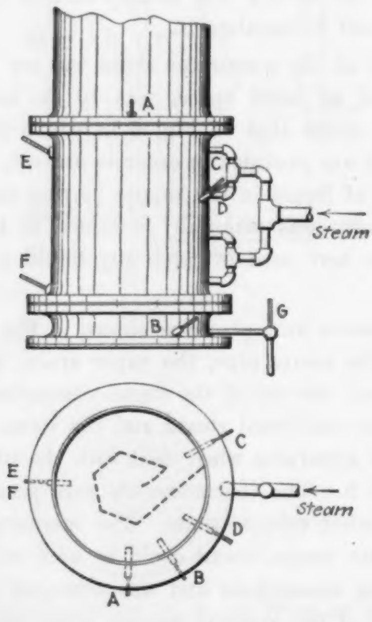



FIG. 8 DIAGRAM SHOWING LOCATION OF THERMOMETERS CALANDRIA E

test ended with the same level. If the boiling was not very rapid and violent the level, as shown by the gage glass, could be read with sufficient accuracy to enable short tests to be run. When there was more rapid boiling tests of longer duration became necessary.

27 Readings of the instruments were taken every 5 minutes throughout the tests.

EFFECT OF HYDROSTATIC HEAD

28 Tests were made on calandrias A, C, D and E in order to secure data on the effect of hydrostatic head. Four series of tests were run under practically the same conditions of steam and boiling pressures. The height of boiling was varied in each series and other conditions kept constant, or as nearly so as possible. Water was used for boiling in these tests. The results are given in Table 2 of the appendix and are plotted in Fig. 9.

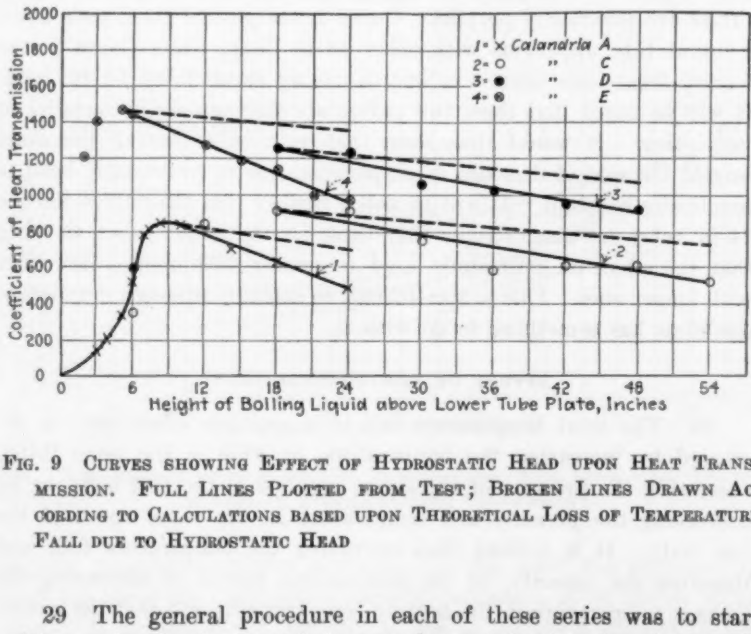



FIG. 9 CURVES SHOWING EFFECT OF HYDROSTATIC HEAD UPON HEAT TRANSMISSION. FULL LINES PLOTTED FROM TEST; BROKEN LINES DRAWN ACCORDING TO CALCULATIONS BASED UPON THEORETICAL LOSS OF TEMPERATURE FALL DUE TO HYDROSTATIC HEAD

29 The general procedure in each of these series was to start with very low heads, a test being run for each head and each succeeding test being made with increased head until the desired maximum was reached. At very low heads only the lower portions of the tubes were kept wet by the boiling water, hence more or less of the heating surface was ineffective, the maximum evaporation being obtained with the head at which the ebullition was sufficient to project the boiling water just to the tops of the tubes. Further increase of head showed a decrease in the rate of evaporation and in the coefficient of heat transmission.

30 The left-hand end of each of the curves plotted in Fig. 9

represents as nearly as can be determined this maximum evaporation. Points obtained with lower heads than this are plotted without drawing curves through them with the exception of Curve 1. The broken lines represent the theoretical decrease in heat transmission due to increase in head, determined as described in Par. 12. It will be noted that the actual curves drop below the theoretical. Just why this is so cannot be determined with certainty, although it may be due to a decrease in the velocity of liquid circulation as the head is increased.

31 Curves 1 and 4, Fig. 9, were both obtained from tests on calandrias with tubes 24 in. long. It will be noted that these two curves are practically parallel. Curve 2 was plotted from tests upon a liquid tube apparatus with tubes 48 in. long, while Curve 3 was plotted from tests upon a calandria having steam tubes 54 in. long. It will be noted that these two curves are also practically parallel to each other. It would thus seem that with tubes having the same length, the rate of decrease in evaporation due to hydrostatic head is practically constant. It will be noted further that the curves for the 24-in. tubes are much steeper than those for the longer tubes, showing that the effect of hydrostatic head is greater with short tubes than with longer ones. This is also difficult to explain, although circulation doubtless has something to do with it.

### EFFECT OF TEMPERATURE LEVEL

32 The total temperature fall in a multiple effect may be increased by increasing the temperature, or what is the same thing, increasing the pressure of the steam supplied to the first body, or by decreasing the pressure and temperature in the vapor space of the last body. It is evident that increasing the temperature fall, and therefore the capacity, by the first method results in increasing the average temperature in the heating compartments and that the second method results in decreasing it.

33 The relative advantages of high steam pressure in the first body as compared with high vacuum in the last body in obtaining increased heat transmission has been a matter of some question by many. Then, too, the reason for the inequality of temperature fall in the different bodies of a multiple effect, the greatest fall being always in the last body where the temperature level is lowest, is not definitely known, although it has been thought that the lowest steam density in the last body might be partly responsible.

34 In order to get data that would aid in settling these points

a large number of tests were made which are given in Table 3. These tests consist of five series B-1 to B-5 inclusive, each series being made upon a different calandria, but in which the temperature level was varied in practically the same manner, all other conditions in each series being maintained as nearly constant as was possible. The limits of temperature in calandria, although varying somewhat in the different series, approximate those with which multiple effects are operated in practice.

35 The relations between temperature of steam in calandria and coefficient of heat transmission are shown graphically in the curves of Fig. 10. The downward trend of the curves in the low temperature

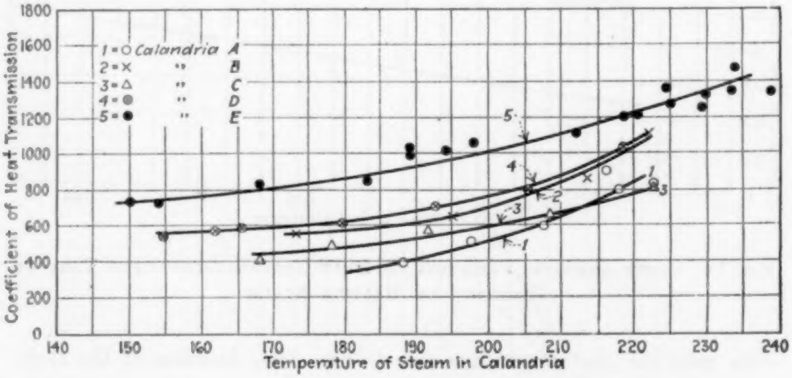



FIG. 10 CURVES SHOWING EFFECT OF TEMPERATURE LEVEL UPON HEAT TRANSMISSION IN FIVE DIFFERENT CALANDRIAS

regions gives unmistakable evidence that the lower the temperature level the lower the coefficient of heat transmission, other things being equal. It is most likely that this is due to the lower density of the steam average. Fig. 11 gives an average of the five curves of Fig. 10, but with steam densities instead of temperatures as abscissas. Some of the falling off in capacity at the low pressures may be due to air in the steam. There would naturally be more leakage as the vacuum increased as well as a greater proportion of air to steam. All of the tests were made, however, with tight joints and with the observed temperatures of steam in calandria indicating no partial air pressure. The variation in height of the curves in Fig. 10 is due to various reasons which will be discussed later.

## EFFECT OF INCONDENSABLE GASES ON HEAT TRANSMISSION

36 For the purpose of studying the influence of incondensable gases a series of tests was made on calandria A, Fig. 3. The amount of air present in the heating steam was regulated by varying the speed of the condensate pump and by admitting air into the steam compartment through a pet cock shown in Fig. 1. The quantity of air present was determined by the temperature method. Through actual observation of the temperatures in the steam compartment the partial steam pressure was obtained and this subtracted from the gage pres-

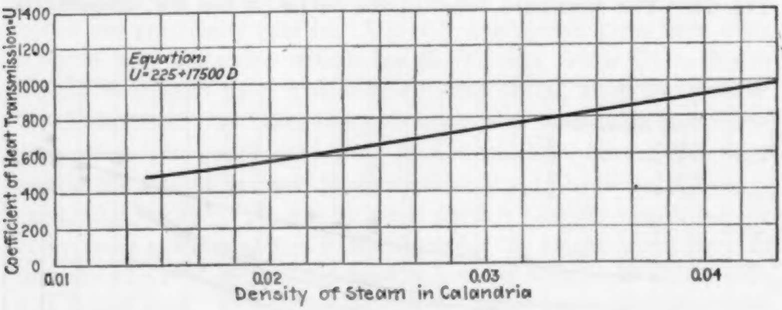



FIG. 11 CURVE SHOWING VARIATION IN HEAT TRANSMISSION WITH VARYING DENSITIES OF HEATING STEAM

sure gave the partial pressure due to air. The location of the thermometers used for measuring the temperatures in the steam compartment are shown in Fig. 1, two thermometers being used, one near the bottom and one near the top, both of which were about 45 deg. from the left steam pipe entrance. Thermometer wells containing mercury and extending about 2 in. into the steam space were used. The average reading of these two thermometers was assumed to be the temperature in the steam compartment.

37 The results of these tests are given in Table 4 of the appendix. Fig. 12 shows the results graphically, $\frac{P_s}{P}$ being the ratio of the partial steam pressure to the observed gage pressure. In the equation for the curve

$$U = \left(\frac{P_s}{P_t}\right)^n$$

the exponent $n$ has a value of nearly 3, showing only a slightly greater decrease of the coefficient due to air than in the tests by George A. Orrok[1] on the effect of air in surface condensation in which the value

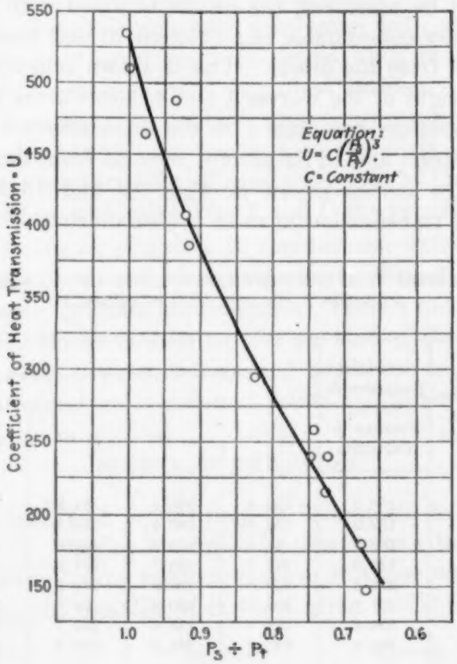



FIG. 12 CURVE SHOWING EFFECT OF AIR UPON HEAT TRANSMISSION

of 2 was given to $n$. The absolute pressure in the evaporator tests was about 16.5 in. of mercury, whereas in the Orrok tests it was only about 2 in. These evaporator tests differed also from the condenser tests referred to as regards hydrostatic head and the range of the values of the coefficient of heat transmission. In the Orrok tests the maximum coefficient was only slightly above 300 while in the evaporator tests it was above 500. This latter was probably due to the greater steam density in the evaporator tests.

38 Attention has already been called to the fact that the evap-

[1]Air in Surface Condensation, Geo. A. Orrok, Trans.Am.Soc.M.E., vol. 34, p. 713.

orators shown in Figs. 6 and 7 were designed for the especial purpose of overcoming the bad effects of incondensable gases. While it was practically impossible to determine the amount of air present, for the reason that the temperature existing adjacent to the heating surface could not be measured, the results obtained with these two arrangements show considerable improvement in heat transmission over those obtained from the others. This is shown roughly in Fig. 9 by the relative height of the Curves 1 and 4, also Curves 2 and 3. The increase is doubtless due mainly to the more effective separation of air from the steam and to its prompt removal from the heating compartment. The curves shown in Figs. 10 and 15 also show the increased heat transmission in these two calandrias.

TABLE 1 VARIED TEMPERATURE CONDITIONS IN CALANDRIA DUE TO PRESENCE OF AIR AND SUPERHEAT

| Test Number | Absolute Pressure in Calandria, In. Mercury | Saturation Temperature due to Pressure in Calandria | Temperature, Deg. Fahr., Steam in Calandria | | | | |
|---|---|---|---|---|---|---|---|
| | | | at D* | at F* | at E* | at C* | at H* |
| 60 | 7.6 | 150.1 | 150.0 | 150.4 | 151.8 | 151.5 | 147.3 |
| 61 | 7.99 | 152.2 | 151.4 | 156.4 | 153.9 | 154.5 | 148.8 |
| 62 | 11.12 | 166.0 | 165.9 | 172.3 | 167.6 | 170.8 | 160.3 |
| 70 | 34.75 | 219.5 | 221.3 | 220.7 | 221.9 | 219.4 | 215.2 |
| 73 | 39.84 | 226.6 | 229.0 | 229.6 | 230.3 | 230.0 | 222.6 |
| 74 | 40.55 | 227.6 | 230.17 | 230.8 | 228.7 | 231.4 | 223.7 |
| 75 | 32.10 | 230.8 | 234.4 | 233.0 | 234.7 | 232.3 | 226.1 |
| 76 | 43.57 | 231.5 | 235.4 | 233.8 | 235.6 | 234.0 | 227.0 |

* See Figs. 7 and 8.

39 It will be noticed that the steam used in the tests plotted in Fig. 12 was superheated and although the temperatures in the calandria were taken where steam and condensation were in more or less intimate contact the temperatures found may have been slightly above what would have existed if saturated steam had been used. This probably accounts for the fact that the upper end of the curve shows no air, whereas it is reasonable to suppose there was a small amount present. In view of this fact the position of the curve may be slightly inaccurate and its value comes mainly from its direction. In several tests on calandria D and calandria E large amounts of air were admitted through the pet cock used for admitting air in the series plotted in Fig. 12. In fact, judging from the amount the pet

cock was open, more air was admitted than in any of the tests plotted in the series. It was found, however, that such admission of air had little effect in reducing the coefficient of heat transmission.

40 The thermometers inserted in the steam space of calandrias A, B, and C, where temperatures were taken both at the top and bottom, seldom gave equal readings except when there was no air present. This condition occurred only when the observed temperature was equal to or greater than the saturation temperature corresponding to the observed pressure. In the tests on calandria A, the greater temperature was in practically all cases registered by the thermometer at the top, whereas with calandrias B and C the reverse was the case.

41 In the tests on calandria E, thermometer wells were inserted at various parts of the calandria as shown in Figs. 7 and 8 for the purpose of observing steam temperatures. Table 1 shows some interesting data selected at random to give an idea of the unstable conditions due to the presence of air and superheat as shown by the variation of temperature at different points and times.

### DENSITY OF THE LIQUID

42 The density of liquids may be determined by either of two hydrometers, viz., Brix or Beaumé. In these tests a Brix hydrometer was used. The following tabulation shows relations which may be of use in grasping the meaning of the data given in Table 5.

| Brix | Beaumé | Specific Gravity |
|---|---|---|
| 0 | 0 | 1.0 |
| 10 | 5.7 | 1.0401 |
| 20 | 11.3 | 1.0833 |
| 30 | 16.8 | 1.1296 |
| 40 | 22.3 | 1.1794 |
| 50 | 27.7 | 1.2327 |
| 60 | 33.0 | 1.2899 |
| 70 | 38.1 | 1.3509 |

The Brix spindle reads directly the percentage of solids in a solution.

43 Increasing the density of the liquid being boiled decreases the heat transmission by decreasing the temperature fall. The circulation is also affected by the density and this also affects heat transmission. The boiling temperature of solutions increase with the density and in order to get first-hand data on this subject, a series of tests was made upon calandria E, the density of the sugar juice being

varied from 18 to 70 Brix and the height of the juice kept at 12 in. The results of these tests are shown in series D-1, Table 5. By subtracting the saturation temperature corresponding to the pressure on the boiling liquid from the observed temperature of the boiling liquid

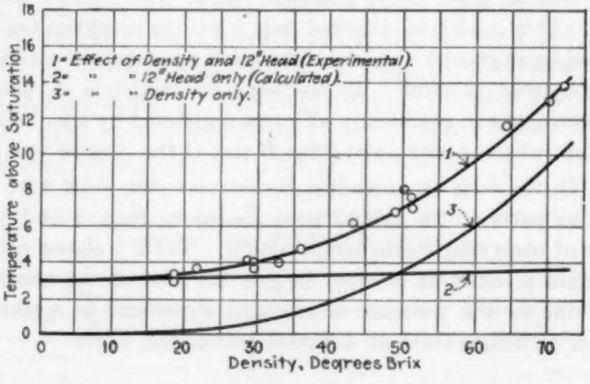



FIG. 13 CURVES SHOWING EFFECT OF DENSITY OF LIQUID UPON BOILING TEMPERATURE

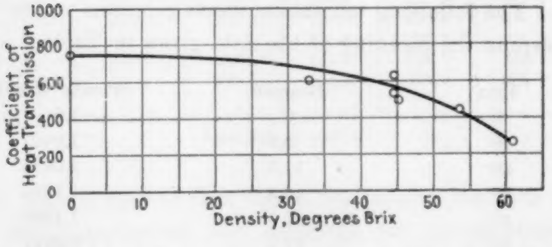



FIG. 14 CURVE SHOWING EFFECT OF DENSITY OF LIQUID UPON HEAT TRANSMISSION

the increase in the boiling temperature due to the combined effect of hydrostatic head and juice density was obtained. Curve 1, Fig. 13, was obtained by plotting these against the corresponding densities. Curve 2 represents the calculated loss in temperature fall due to the 12-in. hydrostatic head for the conditions of the tests. By subtracting the ordinates of Curve 2 from corresponding ones of Curve 1, Curve 3, representing the loss due to density alone, was obtained. The

equation of this curve is $y = CD^{3.1}$ in which $C$ = a constant and $D$ = density of liquid, Brix. These tests were very carefully made after many preliminary tests for the purpose of controlling conditions properly and it is believed that the true relation between density and decrease in temperature fall is shown in Curve 3.

44 Another set of tests was made to determine the actual variation in heat transmission as affected by density. The data from these

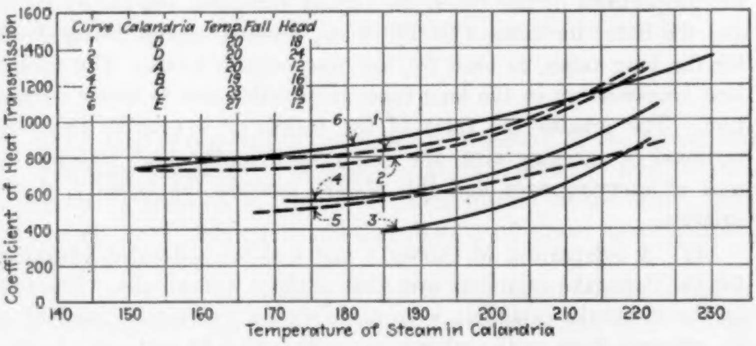



FIG. 15 CURVES COMPARING HEAT TRANSMISSION IN DIFFERENT CALANDRIAS TESTED. FULL LINE CURVES COPIED FROM FIG. 10; BROKEN LINE CURVES SAME AS IN FIG. 10 AFTER CORRECTION FOR HYDROSTATIC HEAD

tests are given in series D-2 of Table 6 and shown graphically in Fig. 14.

### TYPE OF CALANDRIA

45 The five calandrias tested differ from each other in several fundamental features to which attention has already been called. Most important among these differences of design may be mentioned proportions of tubes, that is, ratio of length to diameter; the use or non-use of downtakes to aid the circulation of the juice and the methods employed for removing incondensable gases. What appear to be more or less conclusive data bearing upon the relative merits of these different designs were obtained in the tests. This is probably brought out best in the hydrostatic tests, Fig. 9, also in the temperature level tests, Fig. 10. Both of these sets of curves, however, show the effects of difference of design only in the rough, for the reason that the conditions of operation were not the same in all of the tests. In the

case of the temperature level tests there was considerable variation in the hydrostatic head. In order to make these tests more nearly comparable the curves of Fig. 10 are reproduced in Fig. 15 with the necessary corrections for hydrostatic head based upon the curves of Fig. 9. The full line curves are taken without change from Fig. 10. The broken line curves are those corrected for hydrostatic head. The principal conditions during the tests are noted on the plat.

46 Curves 3 and 5 show something as to the effect of varying the proportions of the tubes, the former for tubes 1⅜ in. by 24 in. and the latter for tubes 2 in. by 48 in., some advantage being shown for the long tubes, at least for low temperature levels. The greater heat transmission in the long tubes is probably due to better circulation. The greater the ratio of the length of a tube to its cross-sectional or carrying area, the greater will be the heat transfer per unit of carrying area and this should increase the velocity of circulation.

47 A comparison of Curves 3 and 4 shows a decided advantage for the downtake calandria over that without a downtake. The tests on the downtake calandria were made with a hydrostatic head of 16 in., whereas those on the other were made with a 12-in. head. Correction for this difference would slightly increase the advantage shown for the downtake calandria.

48 Curves 1 and 6 show the results of the tests on two types of evaporators designed especially for efficient removal of incondensable gases. Both show coefficients of heat transmission considerably in excess of those obtained with the standard types represented by Curves 3, 4 and 5. Curve 6 for calandria E, the baffle plate calandria with a head of 12 in., is only slightly higher than Curve 1 which represents D, the double tube calandria with an 18 in. head. These relations are also corroborated in a general way by the curves of Fig. 9. While the good results shown for these two calandrias may undoubtedly be attributed mainly to the more efficient separation of incondensable gases from steam and its removal, it is probable that the increased velocity of steam is also partly responsible.

## CONCLUSIONS

49 The loss in heat transmission due to hydrostatic head is considerably in excess of the theoretical, and this excess is greater for short tubes than for long tubes. The loss with varying heads, other conditions being equal, varies according to a straight line formula.

50 Other conditions being equal, the lower the "temperature level" the smaller the coefficient of heat transmission, in other words, the lower the temperature and density of the heating steam the smaller the coefficient. The coefficient of heat transmission varies according to the equation

$$U = 2.25 + 17500\,D$$

where $D$ = density of heating steam in pounds per cubic foot.

51 Air or other incondensable gases in the heating steam greatly reduce the heat transmission, even with relatively low vacua. The coefficient of heat transmission varies according to the equation

$$U = C\left(\frac{P_s}{P_t}\right)^3$$

in which $C$ is a constant, $P_s$ the partial pressure of the steam and $P_t$ the total pressure. The presence of "air pockets" may be conveniently determined by means of thermometers in the steam compartment; good circulation and distribution of steam are important in preventing them.

52 Increasing the density of the boiling liquid causes a loss in heat transmission due to the decrease in temperature fall. As the density is increased the boiling temperature increases according to the equation $y = CD^{3.1}$, in which $C$ = a constant and $D$ = density in deg. Brix. The total loss due to the density of the boiling liquid seems to be in excess of that due to loss of temperature fall. This is probably due to lower velocity of circulation.

53 The great temperature fall required in the last body of a multiple evaporator is due to the combined influence of greater amounts of air, steam of lower density, liquid of higher density, also in many cases, more foul heating surfaces, than in preceding bodies.

54 The downtake or circulation tube was shown to increase heat transmission materially. Long tubes give better results as to heat transmission than short tubes due to better circulation.

55 The tests show that the double tube and the baffle plate calandrias gave greatly increased heat transmission as compared with the standard types tested, and indicate that attention to steam distribution and the removal of incondensable gases is very important.

56 In closing, the author wishes to acknowledge the valuable assistance of Messrs. J. A. Gunther and A. J. Isacks, who did most of the testing and calculating. Thanks are due Mr. T. F. Sanborn of New York City who built and furnished the double tube calandria, Fig. 6, and Mr. A. L. Webre of New Orleans, who built and furnished the baffle plate calandria, Fig. 7. Some 350 tests have been made on the apparatus described, though this paper includes only those fulfilling required conditions.

## APPENDIX

57 In the tables which follow will be found average observed and calculated data taken during the tests. In calandrias A, B and C, temperatures in the steam spaces were obtained by means of thermometers placed both at the bottom and at the top, and the item ''observed temperature'' given in the tables is the average readings from these two temperatures.

58 The temperature fall was found by subtracting the observed temperature in the steam space from the saturation temperature corresponding to the absolute pressure of the heating steam when the observed temperature in the calandria was equal to or greater than the saturation temperature corresponding to the pressure. In case the observed temperature in the calandria was less than the saturation temperature the temperature fall was obtained by subtracting the observed temperature in the vapor space from the observed temperature in the calandria. It was thought best to use saturation temperature in the calandria where superheat was shown, as condensation can take place only at saturation temperatures.

59 The coefficient of heat transmission, that is, ''B.t.u. transmitted per square foot of heating surface per degree of temperature fall per hour,'' was obtained by dividing the B.t.u. transmitted per square foot per hour by the temperature fall. Marks and Davis steam tables were used in working up all the tests.

TABLE 2 HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

To determine the Effect of Hydrostatic Head. Water was used in These Tests

| Test Number | Duration of Test, Min. | Absolute Pressures: Vapor Space, In. Mercury | Absolute Pressures: Calandria, In. Mercury | Temperatures, Deg. Fahr.: Steam Pipe | Temperatures, Deg. Fahr.: Vapor Space, Average | Temperatures, Deg. Fahr.: Calandria, Average | Temperatures, Deg. Fahr.: Juice Entering | Height of Boiling above Lower Tube Plate, In. | Weight of Water Evaporated, Lb. | Temperature Fall, Deg. Fahr. | Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr. | Coefficient of Heat Transmission |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Series A-1. Calandria A. No Downtake. 56 Sq. Ft. Heating Surface. Tubes 1⅜ In. by 24 In. | | | | | | | | | | | | |
| 1 | 30 | 26.98 | 33.57 | 221.5 | 207.7 | 219.6 | 81.0 | 3.0 | 37 | 10.0 | 1.32 | 145 |
| 2 | 30 | 26.45 | 33.85 | 221.3 | 205.0 | 218.0 | 82.0 | 4.0 | 68 | 13.0 | 2.45 | 204 |
| 3 | 30 | 26.52 | 33.89 | 225.7 | 206.0 | 219.3 | 82.0 | 5.0 | 114 | 12.2 | 4.07 | 366 |
| 4 | 30 | 25.84 | 33.66 | 227.0 | 206.4 | 218.8 | 84.0 | 6.0 | 149 | 11.4 | 5.32 | 521 |
| 5 | 30 | 27.24 | 33.78 | 230.4 | 207.6 | 218.0 | 82.0 | 14.0 | 194 | 11.0 | 6.93 | 692 |
| 6 | 30 | 26.85 | 33.63 | 229.6 | 206.4 | 218.0 | 82.0 | 18.0 | 184 | 11.4 | 6.57 | 633 |
| 7 | 30 | 27.06 | 34.04 | 227.5 | 206.6 | 218.0 | 83.0 | 14.0 | 150 | 12.0 | 5.36 | 490 |
| 8 | 30 | 27.08 | 33.71 | 229.5 | 206.6 | 218.0 | 82.0 | 7.0 | 228 | 11.4 | 8.14 | 784 |
| 9 | 30 | 27.03 | 33.51 | 230.1 | 207.0 | 217.4 | 81.0 | 10.0 | 214 | 10.1 | 7.64 | 832 |
| 10 | 20 | 26.89 | 33.63 | 230.5 | 206.8 | 217.2 | 81.0 | 8.5 | 148 | 10.4 | 7.93 | 840 |
| 11 | 20 | 27.07 | 33.87 | 229.0 | 207.0 | 217.4 | 81.0 | 6.5 | 147 | 10.4 | 7.87 | 813 |
| Series A-2. Calandria C. No Downtake. 75.4 Sq. Ft. Heating Surface. Tubes 2 In. by 48 In. | | | | | | | | | | | | |
| 12 | 20 | 21.80 | 29.93 | 231.4 | 198.6 | 211.0 | 81.5 | 6.0 | 104 | 12.4 | 4.14 | 365 |
| 13 | 20 | 22.04 | 30.01 | 245.5 | 197.2 | 211.65 | 82.0 | 12.0 | 281 | 14.45 | 11.18 | 844 |
| 14 | 20 | 21.93 | 29.81 | 249.4 | 196.8 | 211.7 | 81.0 | 18.0 | 310 | 14.9 | 12.3 | 908 |
| 15 | 20 | 21.96 | 29.93 | 248.7 | 196.8 | 212.15 | 81.0 | 24.0 | 318 | 15.21 | 12.7 | 912 |
| 16 | 30 | 21.76 | 29.90 | 240.0 | 196.4 | 210.8 | 82.5 | 30.0 | 368 | 14.4 | 9.76 | 741 |
| 17 | 30 | 22.21 | 30.04 | 240.8 | 196.6 | 210.9 | 82.8 | 36.0 | 284 | 14.3 | 7.54 | 576 |
| 18 | 30 | 21.74 | 29.90 | 236.1 | 196.2 | 210.4 | 81.5 | 42.0 | 306 | 14.2 | 8.12 | 605 |
| 19 | 30 | 21.80 | 29.88 | 240.8 | 195.6 | 210.85 | 89.0 | 48.0 | 323 | 15.25 | 8.57 | 609 |
| 20 | 30 | 21.91 | 29.92 | 238.6 | 196.1 | 211.15 | 85.0 | 54.0 | 270 | 15.05 | 7.2 | 517 |

TABLE 2—*Continued*

| Test Number | Duration of Test, Min. | Absolute Pressures | | Temperatures, Deg. Fahr. | | | | Height of Boiling above Lower Tube Plate, In. | Weight of Water Evaporated, Lb. | Temperature Fall, Deg. Fahr. | Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr. | Coefficient of Heat Transmission |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Vapor Space, In. Mercury | Calandria, In. Mercury | Steam Pipe | Vapor Space, Average | Calandria, Average | Juice Entering | | | | | |
| Series A-3<br>Calandria D. 86.21 Sq. Ft. Heating Surface<br>Tubes 2 In. by 54 In. | | | | | | | | | | | | |
| 21 | 23 | 20.46 | 29.75 | 241.7 | 194.8 | 219.2 | 70.0 | 6.0 | 300 | 16.8 | 11.8 | 593 |
| 22 | 42 | 22.05 | 30.22 | 228.0 | 198.2 | 212.5 | 70.0 | 12.0 | 940 | 14.15 | 15.23 | 1196 |
| 23 | 41 | 22.25 | 30.26 | 254.0 | 198.1 | 212.4 | 70.0 | 18.0 | 936 | 14.3 | 15.92 | 1257 |
| 24 | 27 | 21.45 | 29.80 | 257.2 | 196.1 | 210.2 | 70.0 | 24.0 | 606 | 14.1 | 15.65 | 1235 |
| 25 | 48 | 21.95 | 29.83 | 232.0 | 197.0 | 211.5 | 71.0 | 30.0 | 640 | 14.5 | 13.65 | 1048 |
| 26 | 43 | 21.56 | 30.10 | 255.2 | 195.8 | 211.5 | 71.0 | 36.0 | 894 | 15.7 | 14.49 | 1025 |
| 27 | 16 | 21.54 | 29.13 | 247.3 | 195.1 | 210.7 | 70.0 | 42.0 | 305 | 15.55 | 13.28 | 943 |
| 28 | 15 | 21.68 | 29.38 | 244.4 | 196.5 | 214.3 | 70.0 | 48.0 | 305 | 15.1 | 14.15 | 912 |
| Series A-4<br>Calandria E. 29 S.. Ft. Heating Surface<br>Tubes 1¾ In. by 24 In. | | | | | | | | | | | | |
| 29 | 20 | 21.74 | 31.71 | ..... | 196.8 | 216.5 | 80.0 | 2.0 | 193 | 18.0 | 19.98 | 1220 |
| 30 | 20 | 23.05 | 33.18 | ..... | 199.0 | 219.2 | 78.0 | 3.0 | 226 | 17.9 | 23.45 | 1410 |
| 31 | 20 | 22.87 | 32.23 | ..... | 198.8 | 220.2 | 78.0 | 5.0 | 228 | 16.8 | 23.55 | 1468 |
| 32 | 20 | 22.53 | 31.61 | ..... | 198.6 | 219.0 | 78.0 | 6.0 | 214 | 16.0 | 22.14 | 1445 |
| 33 | 20 | 22.50 | 39.82 | ..... | 198.7 | 229.7 | 79.5 | 12.0 | 312 | 28.0 | 32.25 | 1256 |
| 34 | 20 | 23.65 | 35.78 | ..... | 199.8 | 225.0 | 79.0 | 15.0 | 226 | 21.2 | 23.4 | 1192 |
| 35 | 30 | 22.81 | 35.62 | ..... | 199.1 | 225.8 | 79.5 | 18.0 | 337 | 21.85 | 23.22 | 1120 |
| 36 | 30 | 22.94 | 35.48 | ..... | 200.4 | 223.3 | 78.0 | 21.0 | 270 | 20.1 | 18.63 | 993 |
| 37 | 20 | 22.92 | 32.52 | ..... | 197.4 | 220.6 | 78.0 | 24.0 | 166 | 18.6 | 17.17 | 974 |

TABLE 3 TESTS OF HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

To determine the Effect of Varying the Temperature Level. Water was used in These Tests

| Test Number | Duration of Test, Min. | Absolute Pressures | | Temperatures, Deg. Fahr. | | | Weight of Water Evaporated, Lb. | Temperature Fall, Deg. Fahr. | Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr. | Coefficient of Heat Transmission |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Vapor Space, In. Mercury | Calandria, In. Mercury | Steam Pipe | Calandria, Average | Juice Entering | | | | |
| Series B-1<br>Calandria A. Without Downtake. 56 Sq. Ft. Heating Surface. Head 12 In.<br>Tubes 1¾ In. by 24 In. | | | | | | | | | | |
| 38 | 20 | 23.72 | 35.93 | 235.0 | 222.45 | 82.0 | 303 | 21.27 | 16.23 | 835 |
| 39 | 20 | 22.52 | 34.09 | 242.9 | 218.0 | 81.0 | 279 | 20.5 | 14.94 | 800 |
| 40 | 20 | 21.01 | 31.97 | 243.8 | 215.3 | 82.0 | 263 | 19.42 | 14.08 | 913 |
| 41 | 20 | 18.27 | 27.59 | 246.0 | 207.7 | 80.0 | 202 | 19.7 | 10.82 | 601 |
| 42 | 20 | 14.77 | 22.64 | 216.3 | 197.4 | 80.5 | 177 | 20.0 | 9.48 | 517 |
| 43 | 20 | 12.11 | 19.14 | 226.6 | 188.1 | 82.0 | 141 | 20.45 | 7.55 | 399 |
| Series B-2<br>Calandria B. With Downtake. 53.26 Sq. Ft. Heating Surface. Head 16 In.<br>Tubes 1¾ In. by 24 In. | | | | | | | | | | |
| 44 | 60 | 29.84 | 35.93 | 241.2 | 222.0 | 83.0 | 494 | 9.2 | 9.28 | 1108 |
| 45 | 60 | 21.44 | 29.93 | 242.6 | 213.7 | 83.0 | 685 | 16.4 | 12.87 | 857 |
| 46 | 60 | 14.11 | 21.46 | 240.1 | 195.1 | 81.0 | 671 | 21.4 | 12.60 | 647 |
| 47 | 60 | 8.09 | 14.02 | 237.9 | 172.8 | 81.0 | 671 | 24.3 | 12.60 | 558 |
| Series B-3<br>Calandria C. 75.4 Sq. Ft. Heating Surface. Head 24 In.<br>Tubes 2 In. by 48 In. | | | | | | | | | | |
| 48 | 20 | 7.00 | 12.48 | 225.3 | 168.1 | 81.5 | 245 | 26.0 | 9.75 | 404 |
| 49 | 20 | 9.30 | 14.61 | 233.6 | 178.2 | 80.0 | 247 | 21.9 | 9.85 | 487 |
| 50 | 20 | 12.40 | 19.40 | 244.0 | 191.6 | 80.5 | 283 | 21.1 | 16.22 | 578 |
| 51 | 20 | 19.26 | 27.53 | 252.3 | 208.8 | 81.5 | 312 | 16.3 | 12.46 | 660 |
| 52 | 20 | 23.88 | 36.04 | 259.2 | 222.5 | 82.0 | 388 | 21.0 | 15.44 | 806 |

TABLE 3—*Continued*

| Test No. | Duration of Test, Min. | Absolute Pressures: Vapor Space, In. Mercury | Absolute Pressures: Calandria, In. Mercury | Temperatures, Deg. Fahr.: Steam Pipe | Temperatures, Deg. Fahr.: Calandria, Average | Temperatures, Deg. Fahr.: Juice Entering | Weight of Water Evaporated, Lb. | Temperature Fall, Deg. Fahr. | Water Evaporated per Sq. Ft. Heating Surface, Lb. per Hr. | Coefficient of Heat Transmission |
|---|---|---|---|---|---|---|---|---|---|---|
| **Series B-4. Calandria D. 86.21 Sq. Ft. Heating Surface. Head 36 In. Tubes 2 In. by 54 In.** | | | | | | | | | | |
| 53 | 25 | 22.14 | 33.72 | 243.9 | 218.5 | 76.1 | 678 | 20.0 | 18.9 | 1029 |
| 54 | 20 | 17.27 | 26.25 | 245.0 | 205.3 | 76.1 | 428 | 20.45 | 14.85 | 797 |
| 55 | 20 | 12.94 | 19.8 | 229.2 | 192.4 | 76.1 | 395 | 19.8 | 13.74 | 700 |
| 56 | 20 | 9.45 | 14.92 | 230.8 | 179.7 | 76.1 | 225 | 20.1 | 11.65 | 628 |
| 57 | 30 | 6.25 | 10.81 | 239.1 | 165.5 | 73.5 | 551 | 23.3 | 12.84 | 594 |
| 58 | 36 | 5.64 | 9.94 | 236.4 | 161.7 | 73.5 | 545 | 21.25 | 12.65 | 583 |
| 59 | 20 | 5.30 | 8.38 | 229.4 | 154.8 | 76 0 | 273 | 19.4 | 9.5 | 527 |
| **Series B-5. Calandria E. 29.0 Sq. Ft. Heating Surface. Head 12 In. Tubes 1¾ In. by 24 In.** | | | | | | | | | | |
| 60 | 30 | 4.1 | 7.6 | ..... | 150.9 | 79.0 | 224 | 22.44 | 15.45 | 738 |
| 61 | 30 | 4.11 | 7.99 | ..... | 154.05 | 79.0 | 243 | 24.60 | 16.74 | 725 |
| 62 | 30 | 6.22 | 11.12 | ..... | 169.15 | 79.0 | 261 | 23.20 | 18.05 | 830 |
| 63 | 25 | 8.28 | 15.05 | ..... | 181.6 | 79.0 | 250 | 25.7 | 20.70 | 857 |
| 64 | 25 | 8.04 | 15.84 | ..... | 189.0 | 77.5 | 353 | 28.75 | 29.27 | 1045 |
| 65 | 30 | 8.68 | 16.66 | ..... | 189.4 | 77.0 | 390 | 28.60 | 26.80 | 992 |
| 66 | 30 | 9.67 | 18.62 | ..... | 194.1 | 77.0 | 409 | 29.60 | 28.20 | 1014 |
| 67 | 25 | 10.46 | 20.66 | ..... | 196.6 | 79.0 | 363 | 30.5 | 30.5 | 1056 |
| 68 | 20 | 13.57 | 27.48 | ..... | 212.2 | 78.0 | 327 | 32.28 | 33.8 | 1110 |
| 69 | 25 | 19.28 | 32.27 | ..... | 219.2 | 77.5 | 451 | 24.80 | 27.6 | 1202 |
| 70 | 30 | 23.59 | 34.75 | ..... | 220.8 | 80.0 | 300 | 18.65 | 20.7 | 1215 |
| 71 | 20 | 23.46 | 35.25 | ..... | 224.4 | 79.0 | 255 | 20.40 | 26.2 | 1358 |
| 72 | 25 | 20.78 | 35.43 | ..... | 225.0 | 77.5 | 387 | 26.5 | 31.3 | 1270 |
| 73 | 20 | 22.50 | 39.84 | ..... | 229.7 | 79.5 | 312 | 28.0 | 32.25 | 1256 |
| 74 | 30 | 22.46 | 40.55 | ..... | 230.28 | 80.0 | 511 | 29.15 | 35.27 | 1318 |
| 75 | 30 | 23.01 | 43.10 | ..... | 233.6 | 79.0 | 545 | 30.8 | 37.6 | 1340 |
| 76 | 20 | 23.30 | 43.57 | ..... | 234.7 | 79.0 | 415 | 31.7 | 43.0 | 1488 |
| 77 | 20 | 22.32 | 47.65 | ..... | 238.3 | 78.0 | 670 | 38.2 | 46.25 | 1330 |

TABLE 4 TESTS OF HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

TO DETERMINE THE EFFECT OF AIR IN THE HEATING STEAM. WATER USED IN THESE TESTS

SERIES C

CALANDRIA A. NO DOWNTAKE. 56 SQ. FT. HEATING SURFACE. HEAD 12 IN.

| Test Number | Duration of Test, Min. | Absolute Pressure: Vapor Space, In. Mercury | Absolute Pressure: Calandria, In. Mercury | Temperature, Deg. Fahr.: Saturation Due to Pressure in Calandria | Temperature, Deg. Fahr.: Steam Pipe | Temperature, Deg. Fahr.: Vapor Space | Temperature, Deg. Fahr.: Calandria, Average | Temperature, Deg. Fahr.: Juice Entering | Weight of Water Evaporated, Lb. | Temperature Fall, Deg. Fahr. | Temperature above or below Saturation in Calandria | Water Evaporated per Sq. Ft., Lb. per Hr. | Coefficient of Heat Transmission | $\frac{P_s}{P_t}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 78 | 20 | 8.06 | 17.64 | 186.5 | 244.0 | 151.8 | 186.6 | 80.0 | 357 | 34.8 | 0.1 | 19.1 | 594 | 1.0 |
| 79 | 20 | 7.94 | 17.61 | 186.4 | 242.6 | 149.0 | 186.4 | 80.0 | 345 | 37.4 | 0.0 | 18.5 | 533 | 1.0 |
| 80 | 20 | 7.91 | 17.51 | 186.2 | 238.5 | 151.0 | 186.0 | 80.0 | 311 | 35.0 | − 0.2 | 16.7 | 510 | 0.997 |
| 81 | 20 | 8.08 | 17.56 | 186.3 | 236.1 | 152.2 | 185.0 | 80.0 | 274 | 32.8 | − 1.3 | 14.7 | 464 | 0.973 |
| 82 | 20 | 8.28 | 17.57 | 186.32 | 230.2 | 153.4 | 183.0 | 81.0 | 277 | 29.6 | − 3.32 | 14.85 | 487 | 0.931 |
| 83 | 20 | 8.07 | 17.64 | 186.5 | 231.8 | 152.3 | 182.5 | 80.0 | 242 | 30.2 | − 4.0 | 12.96 | 408 | 0.916 |
| 84 | 20 | 9.03 | 17.62 | 186.45 | 234.7 | 152.2 | 182.3 | 80.0 | 228 | 30.1 | − 4.15 | 12.2 | 386 | 0.913 |
| 85 | 20 | 8.18 | 17.61 | 186.4 | 227.8 | 152.0 | 182.1 | 84.0 | 180 | 30.1 | − 4.3 | 9.64 | 300 | 0.909 |
| 86 | 20 | 8.07 | 17.68 | 186.6 | 225.0 | 154.0 | 177.7 | 81.0 | 167 | 23.7 | − 8.9 | 8.95 | 296 | 0.822 |
| 87 | 20 | 7.82 | 17.57 | 186.32 | 226.8 | 150.3 | 176.7 | 80.0 | 177 | 26.4 | − 9.62 | 9.49 | 284 | 0.811 |
| 88 | 20 | 8.07 | 17.62 | 186.45 | 218.5 | 152.4 | 173.0 | 81.0 | 152 | 20.6 | −13.45 | 8.14 | 258 | 0.742 |
| 89 | 20 | 8.19 | 17.62 | 186.45 | 217.8 | 153.0 | 173.1 | 80.0 | 139 | 20.1 | −13.35 | 7.45 | 241 | 0.744 |
| 90 | 20 | 8.06 | 17.59 | 186.37 | 215.6 | 152.6 | 171.9 | 80.0 | 128 | 19.9 | −14.47 | 6.86 | 215 | 0.724 |
| 91 | 20 | 7.79 | 17.68 | 186.5 | 211.6 | 152.0 | 171.9 | 81.0 | 144 | 19.9 | −14.7 | 7.71 | 240 | 0.721 |
| 92 | 20 | 7.97 | 17.60 | 186.4 | 206.2 | 152.0 | 168.9 | 81.0 | 107 | 16.9 | −17.5 | 5.73 | 179 | 0.676 |
| 93 | 20 | 8.04 | 17.57 | 186.32 | 203.7 | 152.0 | 168.3 | 80.0 | 86 | 15.8 | −18.02 | 4.61 | 147 | 0.668 |

TABLE 5 BOILING TEMPERATURE OF SUGAR JUICE

To determine the Effect of the Density of the Liquid upon Its Boiling Temperature

| Test Number | Absolute Pressure | | Density Juice, Deg. Brix | Temperature, Deg. Fahr. | | | |
|---|---|---|---|---|---|---|---|
| | Vapor Space, In. Mercury | Calandria, In. Mercury | | Boiling Juice | Boiling Corrected to a Pressure of 5.9 In. Mercury | Corresponding to Pressure of Boiling | Boiling above Saturation |
| Series D-1<br>Calandria E. 29 Sq. Ft. Heating Surface. Head 12 In. | | | | | | | |
| 94 | 22.62 | 39.70 | 18.58 | 199.5 | 143.4 | 198.2 | 3.4 |
| 95 | 8.74 | 19.71 | 18.58 | 157.9 | 142.9 | 155.9 | 2.9 |
| 96 | 9.74 | 20.47 | 21.59 | 163.0 | 143.7 | 160.2 | 3.7 |
| 97 | 10.02 | 22.25 | 28.61 | 164.6 | 144.1 | 161.6 | 4.1 |
| 98 | 14.72 | 27.03 | 29.61 | 180.5 | 144.0 | 178.3 | 4.0 |
| 99 | 13.9 | 24.89 | 29.62 | 177.8 | 143.7 | 175.7 | 3.7 |
| 100 | 7.70 | 18.75 | 32.84 | 154.0 | 144.0 | 150.7 | 4.0 |
| 101 | 6.62 | 16.49 | 35.86 | 149.2 | 144.8 | 144.7 | 4.8 |
| 102 | 28.92 | 40.50 | 42.92 | 214.1 | 146.2 | 210.3 | 6.2 |
| 103 | 4.70 | 8.80 | 48.80 | 138.0 | 146.8 | 131.5 | 6.8 |
| 104 | 4.70 | 9.20 | 50.00 | 139.0 | 148.1 | 131.5 | 8.1 |
| 105 | 3.73 | 10.60 | 51.00 | 130.9 | 147.7 | 123.0 | 7.7 |
| 106 | 29.18 | 42.35 | 51.07 | 215.0 | 147.1 | 210.5 | 7.1 |
| 107 | 16.71 | 29.21 | 64.0 | 193.0 | 151.6 | 184.0 | 11.6 |
| 108 | 4.55 | 13.2 | 70.00 | 143.0 | 153.05 | 130.25 | 13.05 |
| 109 | 4.25 | 13.13 | 72.00 | 141.5 | 153.85 | 127.75 | 13.85 |

TABLE 6 TESTS OF HEAT TRANSMISSION THROUGH THE TUBES OF A VACUUM EVAPORATOR

TO DETERMINE THE EFFECT OF VARYING THE DENSITY OF THE BOILING LIQUID UPON THE HEAT TRANSMISSION

| Test Number | Duration of Test, Min. | Absolute Pressure | | Temperature, Deg. Fahr. | | | Density of Juice, Deg. Brix | Weight of Water Evaporated | Coefficient of Heat Transmission |
|---|---|---|---|---|---|---|---|---|---|
| | | Vapor Space, In. Mercury | Calandria, In. Mercury | Vapor Space | Calandria, Average | Juice Entering | | | |
| SERIES D-2<br>CALANDRIA E. 29 SQ. FT. HEATING SURFACE. HEAD 14 IN. | | | | | | | | | |
| | 20 | 5.06 | 8.60 | 134.8 | 156.9 | 81.0 | 0 | 143 | 758 |
| | 25 | 4.93 | 8.74 | 135.07 | 156.1 | 81.0 | 34.1 | 143 | 609 |
| | 25 | 4.85 | 7.90 | 135.6 | 154.6 | 81.0 | 45.42 | 115 | 632 |
| | 20 | 4.89 | 8.44 | 136.2 | 155.2 | 81.0 | 45.42 | 87 | 526 |
| | 20 | 4.77 | 8.27 | 135.2 | 156.6 | 81.0 | 46.16 | 82 | 493 |
| | 20 | 5.00 | 8.72 | 137.7 | 158.0 | 81.0 | 54.08 | 90 | 441 |
| | 30 | 4.65 | 9.53 | 135.1 | 160.6 | 81.0 | 61.86 | 87 | 264 |

## DISCUSSION

H. D. Fisher (written). Some years ago the writer had considerable experience with a calandria of the type shown in Fig. 3.

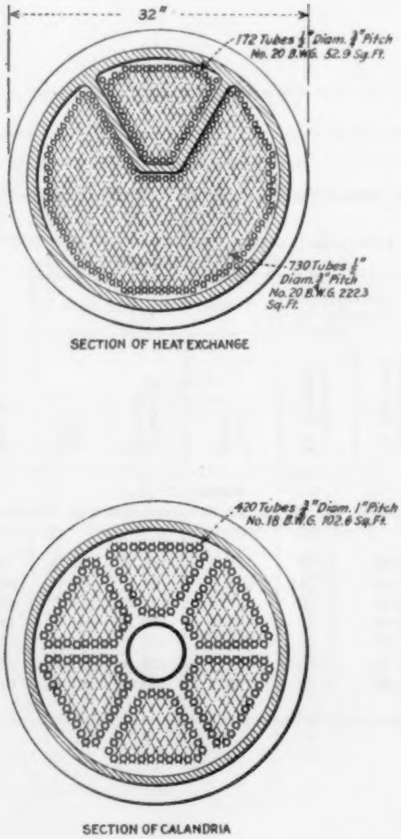



Fig. 16a Experimental Calandria Designed for Testing Out a Compression Cycle of Evaporation

Full details of the apparatus are shown in Figs. 16a and 16b. It was an experimental apparatus designed for testing out a compression cycle of evaporation, in which the vapor of a boiling liquid was compressed until its saturation temperature was high enough to

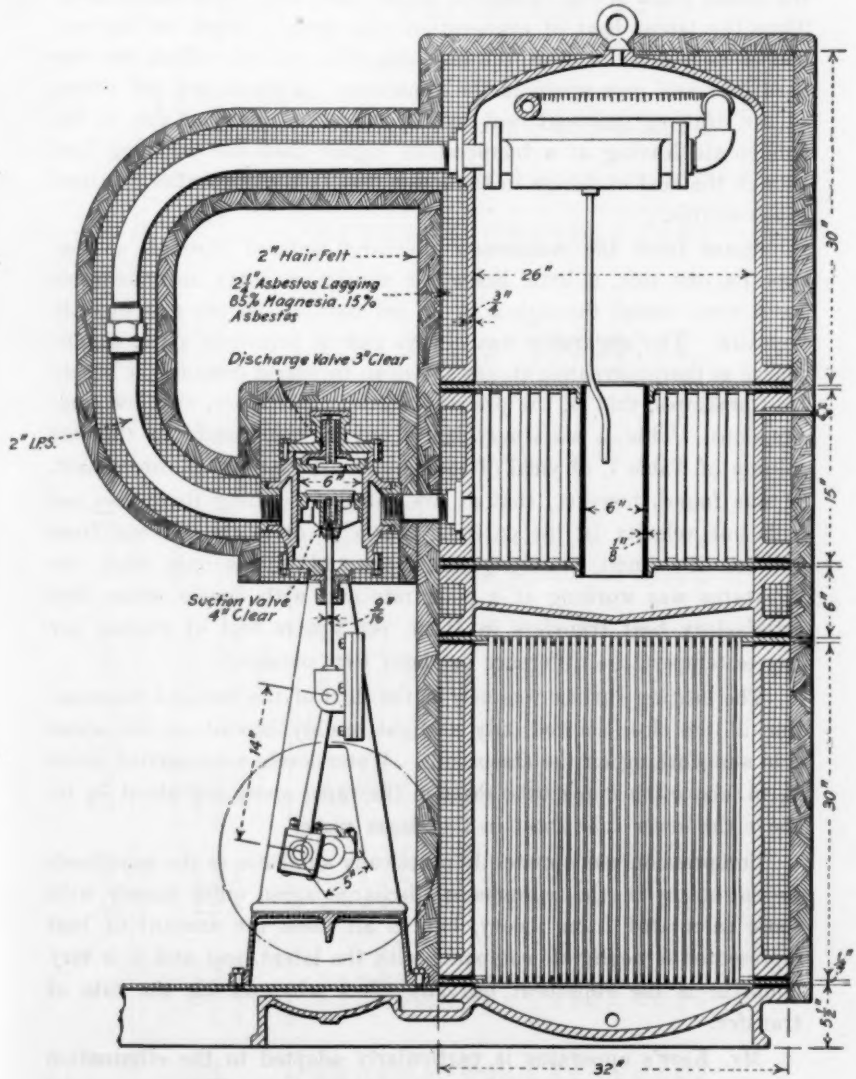



FIG. 16b EXPERIMENTAL CALANDRIA DESIGNED FOR TESTING OUT A COMPRESSION CYCLE OF EVAPORATION

evaporate a corresponding quantity of the liquid when forced into the steam space of the calandria, at the same time condensing itself. Thus the latent heat of evaporation was never rejected to the condenser as it must be from the last body of a multiple effect, but was used over and over again in the apparatus. A steam coil not shown in the drawing supplied heat lost by radiation and that due to the condensate leaving at a temperature higher than the entering feed though the heat exchange in the lower part of the apparatus returned much of this.

Steam from the compresser discharge entered through a 2-in. pipe on one side, a little below the middle, and the incondensable gases were vented through a ⅛-in. pet cock at the top and directly opposite. The apparatus was always run at pressures above atmospheric as thermodynamic theory shows an increased economy at higher base pressures, that is, the pressure in the vapor space, and low compressions. This is strikingly borne out by the results in the last column of Table 7, of yield of condensate per i.h.p-hr. of compressor. It was found, however, that at low rates of running there was not sufficient velocity in the entering steam to drive the air out from between the small, closely-spaced tubes and it was only when the apparatus was working at a good rate and with denser steam that satisfactory heat transfers in B.t.u. per square foot of surface per degree temperature difference per hour were obtained.

The heating surface was new at the start of the test and examination at the close showed only a slight muddy deposit on the water side and nothing on the steam side. Water levels were carried about ¼ in. above the upper tube sheet in the vapor space and about ⅛ in. above the lower tube sheet in the steam space.

Compression was apparently practically adiabatic as the superheats in the steam in the compressor discharge agree quite closely with those calculated from theory, but in all cases the amount of heat represented is negligible compared with the latent heat and it is very doubtful if the superheat has any effect whatever on the rate of transfer.

Mr. Kerr's apparatus is particularly adapted to the elimination of air as he has few large tubes and wide spacings. With commercial designs of calandria even with a steam belt and six or eight contracted openings delivering steam into the tubes at rather high velocities, it is doubtful whether it would be possible to obtain as perfect air

TABLE 7 RESULTS OF TESTS ON FORBES DISTILLING APPARATUS

| Test Number | Duration, Hours | Base Pressure Absolute | Compression Pressure Absolute | Temperature of Saturation, Base Pressure | Temperature of Saturation, Compression Pressure | Temperature of Compressor, Discharge | Superheat, Compressor Discharge | Temperature Difference in Calandria | Weight of Water Distilled, lb. | Heat Transfer, B.t.u. per Sq. Ft. per Deg. Difference per Hour | Yield of Water per I.H.P-Hour lb. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 7.07 | 32.3 | 35.1 | 254.5 | 259.6 | 265.5 | 5.6 | 5.1 | 683.0 | 174 | 317 |
| 2 | 6.13 | 32.3 | 36.6 | 254.5 | 261.9 | 272.9 | 11.0 | 7.4 | 1208.5 | 245 | 190 |
| 3 | 3.37 | 37.5 | 41.4 | 263.4 | 269.4 | 273.1 | 3.7 | 6.0 | 492.5 | 222 | 225 |
| 4 | 3.32 | 41.6 | 46.6 | 269.6 | 276.6 | 279.0 | 2.4 | 7.0 | 549.1 | 214 | 257 |
| 5 | 4.13 | 48.0 | 52.3 | 278.5 | 283.9 | 294.0 | 10.1 | 5.4 | 782.5 | 316 | 284 |
| 6 | 3.19 | 51.0 | 57.3 | 282.3 | 289.7 | 303.1 | 13.4 | 7.4 | 676.9 | 258 | 228 |
| 7 | 3.16 | 51.9 | 56.8 | 283.4 | 289.2 | 292.3 | 3.1 | 5.8 | 594.7 | 291 | 314 |
| 8 | 3.16 | 52.2 | 57.3 | 283.8 | 289.7 | 292.3 | 2.6 | 5.9 | 374.4 | 181 | 253 |
| 9 | 3.42 | 52.4 | 58.6 | 284.0 | 291.2 | 298.1 | 6.9 | 7.2 | 672.4 | 245 | 225 |
| 10 | 3.03 | 57.0 | 62.1 | 289.4 | 295.0 | 301.2 | 6.2 | 5.6 | 430.2 | 227 | 305 |
| 11 | 2.89 | 57.0 | 63.6 | 289.4 | 296.5 | 304.5 | 8.0 | 7.1 | 629.4 | 274 | 222 |
| 12 | 3.12 | 62.1 | 66.9 | 295.0 | 299.9 | 301.7 | 1.8 | 4.9 | 680.7 | 305 | 283 |
| 13 | 3.16 | 62.2 | 68.9 | 295.1 | 301.9 | 310.0 | 8.1 | 6.8 | 816.2 | 339 | 221 |
| 14 | 3.19 | 62.8 | 68.5 | 295.7 | 301.5 | 306.7 | 5.2 | 5.8 | 806.5 | 388 | 242 |
| 15 | 3.18 | 70.8 | 76.0 | 303.7 | 308.5 | 308.5 | 0.0 | 4.8 | 564.4 | 327 | 291 |
| 16 | 3.10 | 71.3 | 79.0 | 304.2 | 311.2 | 321.4 | 10.2 | 7.0 | 852.8 | 349 | 194 |
| 17 | 3.18 | 72.1 | 80.6 | 305.0 | 312.6 | 315.3 | 2.7 | 7.6 | 634.8 | 232 | 207 |
| 18 | 3.20 | 82.1 | 88.3 | 313.9 | 319.0 | 323.2 | 4.2 | 5.1 | 631.8 | 340 | 312 |
| 19 | 3.14 | 82.3 | 90.9 | 314.0 | 321.0 | 327.0 | 6.0 | 7.0 | 1024.3 | 411 | 264 |

venting and as high coefficients of heat transfer as he has done. The ratio of diameter to length of tubes corresponds to the best late practice and should give very satisfactory circulation.

The only criticism the writer has to offer is regarding the length of the tests, as unless steam pressure is very carefully maintained the same at beginning and end, an uncertainty is introduced and with violent ebullition and circulation, it is difficult to be sure a constant water level is maintained inside the apparatus.

A. L. Webre[1] (written). In interpreting the meaning of the results of Professor Kerr's experiments, several facts must be borne in mind:

*a* The apparatus used was very small as compared with the equipments in practical use today. We know that the relation between a scale model and a full sized unit is problematical. It has been our experience, as a rule, that small vacuum evaporators give much better results than large ones. This is probably due to the more thorough removal of foul gases. It is not practical to obtain a coefficient of transmission greater than 250 in every day work with large units of the types described in the paper as calandrias A, B, and C.

*b* It must also be remembered that the displacement of the condensate pump in his apparatus was many times larger in proportion to the surface in operation than in actual practice. On a 3000-sq. ft. body, it is customary to use a 12-in. single cylinder pump with a piston speed of about 50 ft. per minute. Professor Kerr does not give the size nor speed of this pump, but having seen his apparatus personally, we are confident that the displacement was far in excess of that suggested above. This would naturally purge the calandrias more completely of the foul gases, producing very high rates of transmission.

*c* In all the experiments, the temperature of the feed to the evaporator was very low as compared with the boiling point in the apparatus during the tests. It also has been our experience, confirmed by many tests, that far better results are obtainable when the feed enters at or above

[1]Mechanical Engineer, John H. Murphy Iron Works, New Orleans, La.

the boiling point, especially if it is admitted below the tubes. Those who have had experience in the operation of apparatus of this kind in multiples have, doubtless, noticed that the temperature fall in the first of a triple or quadruple effect is always greater than in the second, in spite of the greater density of the liquor in the second body. This is, doubtless, due to the fact that in the first body, the feed is generally below the boiling point, and in the second considerably above it.

The surface in a multiple effect is designed for evaporating and not for heating. To prove this contention, we have tried to use a single body as a surface condenser with water on the inside of the tubes and steam on the outside, and we secured very poor results. As soon, however, as the liquid begins to boil, there is a remarkable change in the coefficient of transmission. This is due to the slow velocity of circulation when the surface is being used for heating, as compared with the very rapid circulation when it is used for boiling. We understand that it has been established in surface condenser practice that the coefficient of heat transmission is proportional to the square root of the velocity of the condensing water in the tubes. The same law should hold good with evaporator practice.

With slow evaporation, small bubbles of steam cling to the surface actually insulating it from contact with the boiling liquid, and it is only after the circulation is established that these are swept off, so that we can readily understand why higher rates of evaporation give higher coefficients, and long tubes are better than short ones.

In this connection, we believe that we should be very guarded about coming to conclusions as to the relative merits of calandrias equipped with downtakes or without them. This feed temperature problem would naturally affect the calandria without downtake more than the one with a downtake, and might neutralize the apparent advantage shown in Professor Kerr's tests, for it is a fact that there are many effects in Cuba without downtakes giving excellent results, fully as good as those otherwise equipped.

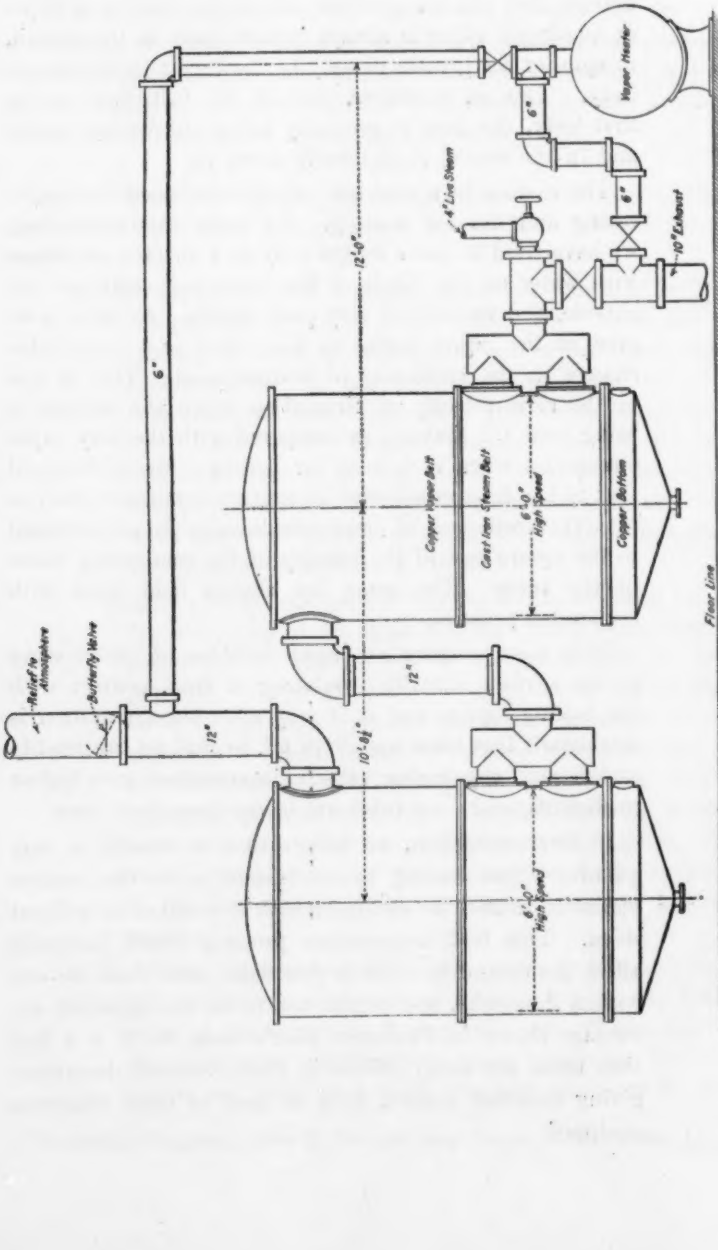



FIG. 17 DESIGN OF ATMOSPHERIC DOUBLE-EFFECT EVAPORATOR WITH VAPOR HEATER

*d* We do not quite agree with Professor Kerr on his comparison of the compromise curves shown in Fig. 15. These curves are made up by altering those shown in Fig. 10. It will be noted that for both calandrias D and E, the two temperature level runs were not made with the levels at the points of maximum evaporation, as shown by the juice level series. These points were 18 in. for D and 5 in. for E. The points selected were 36 in. for D and 12 in. for E.

Professor Kerr's compromise curves were made by selecting 18 in. as the compromise boiling level. Inasmuch as on the level series, D showed a coefficient of 1257 at 18 in. and 1025 at 36 in., the curve was boosted up in the ratio of 1257 to 1025, bringing the said coefficient at exactly the maximum point for D. This is perfectly satisfactory as far as D is concerned. But when the same level is assumed for E and the coefficients are lowered in the ratio of 1256 to 1120, a point nearly 24 per cent below its maximum, we consider that E is then at a disadvantage of 24 per cent as compared with D.

According to Professor Kerr himself, longer tubes give higher coefficients. It is logical to conclude, therefore, that if E had had 54-in. tubes as D had, it would have shown higher rates of transmission than it actually did show, and with a level of 18 in. would have shown its maximum which would have been above the maximum for 24-in. tubes. It seems to us, therefore that in order to give E an even break levels of maximum evaporation should have been selected if any comparison was to be made at all.

In connection with the high rates of evaporation shown by E, which is of our design, we have secured considerably higher coefficients in actual practice. On one of our triples, evaporating 8 lb. per sq. ft. per hour, the drop of temperature in the first body was only 5 deg., and in the second 8 deg. Our ability to boil with a *small drop* has suggested that we operate our effects without vacuum. This we are now doing with remarkable success in atmospheric double-effect evaporators. The capacities are about the same as in the old double effects running under vacuum. Fig. 17 illustrates the arrangement of apparatus for the atmospheric double effect with vapor heater, which has the advantage of doing away with vacuum and water pumps, and the water supply necessary to operate the effects. The pressure in the first calandria is from 6 to 8 lb., which is entirely permissible. The vapors leaving the last body are used for juice

heating, which, of course, leaves the largest part uncondensed. It is our intention to use these vapors to operate a vapor vacuum pan.

We want to extend to Professor Kerr our thanks for the painstaking work which he has done, and the valuable information he has worked up. His tests have established facts, which were known only approximately before, and are very useful in that they point the way to scientific design and operation. Experiments such as he conducted, are impossible on large equipments, and by making proper allowances for this fact, we can interpret their true meaning, and we hope he will incorporate all his data in a good text book, which would be of great service to both manufacturers and operators.

THE AUTHOR. In reply to Mr. Fisher's criticism regarding the short duration of the tests, I will state that the pressure in the heating compartment, also the vacuum in the boiling compartment, was very carefully regulated in all the tests. In fact, they were maintained practically constant throughout each test. Considerable preliminary work was devoted to this matter and the length of the tests was decided upon after it had been shown experimentally that the results were uniform. This is corroborated by the smooth curves shown in the text of the paper. In view of the constant conditions maintained it is believed that the tests were not too short for reasonable accuracy.

The writer agrees in the main with Mr. Webre's observations regarding the fact that the coefficients of heat transmission in the small experimental evaporator used in the tests were greater than are obtained in full sized units; also his reasons therefor, though I will have to take issue with him in some instances. The condensate was removed by a direct-acting 4½ by 6 by 7-in. steam pump, the piston speed being about 40 ft. per min., except in the tests to determine the effect of air where it was varied through wide limits. The displacement of this pump was relatively greater than is generally used in full sized units and this would probably increase the heat transmission somewhat, not only because of a more complete removal of incondensable gases but because of increased steam circulation as well. However, as the speed was kept practically constant for the tests on all the different types of evaporators the results are sufficiently accurate for comparison.

I am inclined to believe that the relatively cool feed would increase rather than decrease the velocity of circulation. Moreover, most of the tests were made with greater rates of evaporation per square foot

of heating surface per hour than are usually obtained in practice, which would mean that the velocities of circulation were also greater. In my opinion one of the principal reasons for the high coefficients of heat transmission in the experimental apparatus was the perfectly clean condition of the heating surface, which cannot be maintained in practice even under the most favorable conditions. There seems to be a progressively increasing fouling of evaporator tubes toward the end of the factory grinding season even when they are boiled out regularly with soda and acid.

Mr. Webre's statement that it is not practicable to obtain a coefficient of transmission greater than 250 in everyday work with large units is correct, though it should be remembered that the coefficient is greater in the first body of a multiple evaporator than in the last body and this figure would represent a good average. The writer has lately made some tests upon a large quadruple film evaporator where transmission coefficients for the four bodies were 545, 648, 600 and 207 respectively, the average being 394. The difference between these coefficients and most of those shown in the paper is not excessive.

I agree with Mr. Webre that the compromise curves of Fig. 15 show the relative heat transmission of the different types only approximately. The juice heads 18 in. for calandria D and 12 in. for E were used instead of those of maximum evaporation because it was thought they would be approximately the same as would be used in practice. With the 24-in. tubes of E, it is quite natural that lower heads can be carried than with the 54-in. tubes of D. It is evident that the calandria with tubes 24 in. long has an advantage over those with tubes 48 in. and 54 in. long as regards juice head; also the heads for maximum evaporation would be lower with the 24-in. tubes. I am inclined, therefore, to think that compromise curves corrected to maximum evaporation heads would not furnish a fair basis of comparison for these two types. Tests recently made on full sized evaporators of types D and E seem to indicate that the ratings shown by the compromise curves are not far from correct.

## No 1414

# THE ART OF ENAMELING OR THE COATING OF STEEL AND IRON WITH GLASS

By Raymond F. Nailler, Elyria, Ohio

Member of the Society

Before going into the technology of this art, it will be interesting to note its history in a general way: where it originated and how it was brought down to the present age. The covering of burnt earthenware, porcelain and some metals with a crude enamel took place about the same time as the discovery of glass. Since the time when man first enameled metals, especially gold, silver and copper, he began to be interested in the problems connected with artistic enamels.

2 Colored enamel earthenware has been found in the ruins of Thebes. In the ruins of a great many other ancient cities of Egypt enameled or glass-covered brickwork has also been discovered. That the Egyptians knew how to adorn silver vessels with enamel pictures has been recorded by Pliny the Elder. From Egypt the enameling arts were transferred to Greece and thence to Rome, and some historians maintain that the enameling art came to Italy by Arabia, Spain and the Balearic Islands, and through Roman expeditions the art passed into England, France and Germany. In the museum at Oxford is an enameled ornament which was found in Somerset, the inscription of which dates to the time of Alfred the Great.

3 As a particular example of early art enameling let us briefly consider the so-called enamel painting. Two periods may here be distinguished: The first, known as the "old Limoges style," was characteristic of the time of Francis I (1515-1547). The enamel plate, generally made of copper, was covered with a dark enamel coat. After firing, figures, weaker or stronger according to the relief desired, were put on with white enamel and the impression of bas-relief thereby brought about. The second period of enamel painting began a generation after the first. This is the so-called period of the "miniature style," which was introduced about the middle of the sixteenth

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

century. It was brought to an extraordinary state of perfection by Jean Petitot (1607-1690).

4 As the demand for the artistic enamels of antiquity and the Middle Ages increased, so the enameling of gold and silver gave way to that of copper vessels, and from this point, if we except the somewhat scattered industry of miniature painting on boxes, such as jewel cases, etc., the absolute decline of the enameling art is to be recorded.

5 In the nineteeth century, which it must be conceded brought one of the most important of Europe's social, educational, and technical revolutions, the enameling business came into great significance and eventually became the enameling industry of today. The commercial introduction of iron and steel made it possible to re-apply the half-forgotten art of enameling iron utensils.

6 In the history of the development of enameling or the modern industrial technology of iron enameling, we are able to speak of two periods with regard to the iron materials employed, viz., of the original enameling of cast iron exclusively and the later application to sheet metal.

7 Before considering the technology of any particular type of enamel it will be well to consider the meaning conveyed by the word enamel. If the present-day enamel be briefly and scientifically designated as a boro-sodium-potassium-aluminum silicate generally colored by metallic oxides, then the following definition given by Popelin deserves to be quoted since this, notwithstanding its length, may be described as most clear and comprehensive: "Enamel is a glass fusible at a low temperature and usually compounded of borates and silicates. This mixture originally colorless combines with the greatest ease with metallic oxides under the influence of a pyrotechnic operation, thereby acquiring various colors according to the nature of the oxide which the enameler can vary at will."

8 The uses to which the general class of ceramic compounds known as enamel is put are varied. We have noted that enamels are used for artistic or decorative purposes. Parallel to this use we may classify a far more important type of enamels as "commercial," and it is with this type of enamel that we are to concern ourselves. The first class of commercial enamels is that used on cast iron and this classification includes the field of sanitary equipment together with various forms of cast-iron kettles and similar pieces of engineering apparatus. The enameling of sheet steel may be taken as the second class of commercial enamels. The most familiar division of this

class includes cooking-ware enamels. The various forms of so-called agate and granite ware as well as the single colors are made by enameling sheet-steel forms. The second class of steel enameling which has more recently come into industrial importance is the manufacture of enamel signs, and the third field includes the manufacture of heavy equipment for large scale food preparation, the dairy industries, and general chemical operations. The apparatus in this last case may take the form of tanks, kettles, evaporators, pipes, etc.

9 In outlining the technology of the enameling industries as a whole we may include to a certain extent all of the above commercial classes, certain details of which are varied to meet the conditions of cast iron and various types of sheet-steel products. The outline following, however, is characteristic of all classes.

10 The first step to be considered is the preparation of the enamel. The purity of the raw materials to be used in the compounding of an enamel must be certain and in the cases where materials can vary in strength the actual analysis of the substance must be known to assure proper results. In many cases the secret of an enamel lies in its formula and so the compounding of the batches is very carefully guarded, only a person of responsibility having charge of weighing the ingredients. The various materials are generally kept in bins which are numbered, the person in charge of the mixing having the formula stated in terms of these numbers. The properly weighed batch is thoroughly mixed, this being accomplished either by shoveling carefully on a specially prepared floor or by mechanical mixing by means of a rotating agitator.

11 The product of the mixing room is taken to the smelter in which the various ingredients are fused together in the form of a mass having the characteristics of glass. The furnace in which this operation takes place is a special reverberatory furnace similar to that used in the puddling process in the manufacture of wrought iron. The coal is placed on a grate at the front end of the furnace and the burning gases pass over the bridge wall, strike the roof, and are deflected against the batch. The first change which may be noted in the smelter is the driving off of the water from the borax which produces a swelling up of the mixture. The fluxing materials then melt down and as the heating is continued, dissolve the more refractory constituents. If the batch is properly stirred and the temperature of the furnace carefully regulated, the product of this operation is a clear transparent glass containing no particles of unfused material.

When a test of the fusion comes up to this standard the furnace is tapped and as the liquid glass flows into a tank of cold water it is broken up by the chilling action of the water, the result being a thorough granulation of the product. The material thus prepared is known as "frit."

12 The next step in the preparation of the enamel involves grinding the frit to a fineness at which it can be applied to the surface of the metal. The mill here used is the ordinary pebble mill lined with porcelain brick and containing very hard flint pebbles. In the cast-iron industry two general processes are used, the dry and the wet, in sheet-steel work only the wet. In the first case the frit is ground dry, the pulverized enamel being sprinkled on the hot cast-iron piece. In the wet process a certain percentage of pure white clay is placed in the mill with the frit and a certain definite amount of distilled water. When certain grades of enamel are to be manufactured various compounds are also added here to aid in producing desired colors, gloss and opacity. An instance of this is the addition of tin oxide in the mill in the production of white cooking utensil enamels. The fineness to which the enamel is ground depends upon the particular use to which it is to be put, manner of application and other factors. In the wet process the essential feature is that it be fine enough to remain suspended in the water assisted by the clay and the so-called "vehicles" mentioned below.

13 At this point we may well take up the consideration of the construction and preparation of the material to which the enamel is to be applied. Cast-iron enamels are applied to castings of the desired shape. In the case of cooking-ware enamels the shapes are constructed by pressing and spinning. For heavier equipment such as jacketed kettles the apparatus is constructed by riveting or preferably welding the sheet steel in the desired form. The selection of the steel with a view to its chemical analysis is of prime importance. A reliable specification reads as follows:

| | |
|---|---|
| Sulphur | below 0.040 |
| Phosphorus | below 0.030 |
| Manganese | about 0.40 |
| Silicon | about 0.010 |
| Carbon | about 0.10 |

The steel must necessarily be free from laminations or other mechanical imperfections.

14 With reference to the construction of heavy apparatus the

sheet ranges from $^3/_{16}$ in. to $3/8$ in. in thickness. Two general methods of construction are in use: The first involves the formation of unit sections fitted with flanges. These sections are enameled separately, bolted together at the flanges and the desired apparatus so constructed. The preferable practice, however, is to construct the apparatus in one piece by means of autogenous welding, thus avoiding the use of gaskets or other packing materials in erection. That the enamel may properly adhere to the surface of the metal, the latter must be free from dirt or scale and in the case of welded joints the welds must have a preliminary grinding to reduce the roughness. The entire surface of the apparatus is then cleaned by pickling or sand blasting, the latter process being altogether used in cleaning large apparatus. As the crude ware leaves the sand blast it has a roughened, clean, metallic surface and is in the proper condition to receive the enamel.

15 Before applying the enamel to the metallic surface it is prepared by a process known as "setting up." This involves the addition of certain chemicals to the enamel as taken from the mill, the function of this addition being to assist the clay in holding the enamel particles in suspension. Substances so added are termed "vehicles." At this stage the enamel must be diluted with distilled water to the proper consistency for application.

16 In applying the enamel to the metallic surface three general methods are in use: The first, applicable to small pieces only, is known as "dipping," the piece being dipped into the enamel, the excess of which is shaken off leaving a thin coating on the metal. The second method, known as "slushing," involves pouring the prepared enamel over the surface and allowing it to drain. The third method, which is the principal one used on larger apparatus, involves spraying the finely ground enamel on the metallic surface by means of the compressed air atomizer.

17 Preliminary to the consideration of firing the enamel we may well review the types of furnaces in use. The first type is known as the muffle furnace and involves the use of a large fire clay oven externally heated by means of coal, gas or other fuel. The apparatus to be fired is placed on suitable supports in this muffle. The other type of furnace is known as the direct-fire furnace, in which the heat from the fire is taken up by the walls of the firing chamber and radiated to the apparatus placed within the chamber on suitable racks. This general class of furnace has two special divisions in that,

on the one hand, the piece is rotated within the furnace, while on the other hand, the piece is allowed to remain stationary. For small work the muffle is in general use, but for the production of large apparatus the direct-fire furnace is necessary. At first thought, the muffle furnace may be considered to have an advantage in that the products of combustion together with the dust from the fire cannot come in contact with the enamel. Again it may seem that a more even heat can be realized in the muffle. On the other hand, with a properly designed direct-fire furnace in which the combustion is complete before the gases reach the firing chamber no trouble is experienced due to their presence or to the dust from the fire. The use of natural gas further does away with this latter possibility.

18 In a furnace for firing smaller ware the charging mechanism is a fairly simple matter, it being necessary merely to place the material in the furnace by means of a small fork operated by hand or mechanically. But in the manufacture of engineering apparatus where a single piece may weigh 3000 to 4000 lb. it is necessary to have a large mechanical charging machine on which the piece may be placed outside the furnace, the arm of the machine then properly placing it in the furnace. The general design of such a machine suggests the charger used in open-hearth practice. The apparatus to which the enamel has been properly applied is placed in the furnace which is maintained at the proper temperature. This temperature varies with the nature of the enamel and in cases of high silicon acid-proof enamels reaches in the neighborhood of 2500 deg. fahr. The control of the burning is made possible by the changes which occur in appearance of the enameled surface as fusion takes place. At first the fine particles of enamel begin to fuse together and a general blistered condition exists, giving the surface a very dull appearance. But as the enamel matures, this dull appearance gives way to a bright glass, which when properly developed over the entire surface is an indication that the piece should be withdrawn from the furnace. The time required to burn a piece properly depends upon the temperature of furnace, thickness of metal and nature of enamel.

19 Nothing has been said so far as to the composition of the enamel or of the number of coats applied. In general there are two kinds of enamel, known as ground coats and cover coats. The former serves as a bond between the enamel and the steel, and the latter serves to build up the body of the enamel and presents the finished surface. In the ground coat color is no object. Its composition is such as to

render it adherent and strong. In addition to the ordinary components cobalt oxide seems to be essential to the production of adherence. The explanation of this is debatable. The cover coat is the one which forms the major part of the enamel and if definite color, opacity, etc., are objects the necessary ingredients for their production are introduced here, assisted by mill additions as noted above. In the manufacture of acid-proof enamel, the cover coat is essentially a high silicate and must be free from any metallic oxides, such as oxide of tin, lead, iron, etc. The piece to be enameled receives one ground coat which is burnt well into the steel at a high temperature. The cover coats may be one or two in number for ordinary enameling, but should be at least triple for acid-proof work. In the production of acid-proof apparatus, the use of a ground coat or cover coat is now eliminated and the same material which in place of being an enamel, as commonly termed, is in reality a boro-silicon glass and without the use of any metallic oxides in its compounding.

20 It is very interesting to note the chemical changes which take place in the various stages of the production of the finished enamel. Avoiding so far as possible deep technicality, they may be summed up as below. As a starting point let us select a cover coat formula used in the production of a "dark blue" cooking ware enamel.

| | | | |
|---|---|---|---|
| Feldspar, lb. | 120 | Saltpeter, lb. | 7 |
| Quartz, lb. | 72 | Oxide of Cobalt, lb. | 7½ |
| Borax, lb. | 80 | Oxide of Manganese, lb. | 1 |
| Cryolite, lb. | 30 | Clay in the mill, per cent | 4 |

First considering the behavior of each of the constituents when heated we can later clearly note the general reactions which take place when they are fused together in the smelter.

21 Feldspar varies greatly in composition but as commonly used in enamels is an aluminum-sodium-potassium silicate of approximately the following analysis:

| | Per Cent |
|---|---|
| Silica ($SiO_2$) | 70 |
| Alumina ($Al_2O_3$) | 17 |
| Soda ($Na_2O$) | 7 |
| Potash ($K_2O$) | 6 |

On smelting, none of these constituents vaporize so we may consider them all later in the smelting of the above enamel.

22 Quartz introduced as a pure glass sand contains practically 100 per cent silica ($SiO_2$).

23 Borax is chemically known as sodium tetra-borate and in the crystalline form as used has a certain definite amount of so-called water of crystallization which must be taken into account. The formula $Na_2B_4O_7.10H_2O$ is resolved into $Na_2O.2B_2O_3.10H_2O$. Without going through the calculations assume this to correspond approximately to the following analysis:

| | Per Cent |
|---|---|
| Water ($H_2O$) | 16.0 |
| Boric Oxide ($B_2O_3$) | 37.0 |
| Soda ($Na_2O$) | 47.0 |

Of these the last two do not vaporize on smelting, but the water is evaporated, hence 100 lb. of borax smelts to 84 lb. of the remaining oxides.

24 Cryolite is a double fluoride of sodium and aluminum, the formula of which may be written $Na_3AlF_6$. When smelted, the sodium and aluminum appear as oxides and from 100 lb. of cryolite we realize about 24 lb. of alumina ($Al_2O_3$), 44 lb. of soda ($Na_2O$) and 54 lb. of fluorine ($F_2$). There is some dispute as to whether or not the fluorine is vaporized. In the present discussion it matters not, hence we shall consider that the third of these three compounds is lost in smelting.

25 Saltpeter is potassium nitrate $KNO_3$. When heated under the conditions of smelting it may be considered to break up into potash ($K_2O$) and nitrogen pentoxide ($N_2O_5$). The reaction is $2KNO_3 = K_2O + N_2O_5$. By calculations based on this reaction 100 lb. of saltpeter yields about 47 lb. of potash ($K_2O$) and 53 lb. of the nitrogen oxide ($N_2O_5$). The latter may be considered as completely vaporized.

26 Oxide of cobalt ($CoO$) may be taken as non-volatile and pure. The same may be assumed for the oxide of manganese ($MnO_2$).

27 To sum up the above as an outline of the reactions taking place in the smelter

| | | |
|---|---|---|
| 120 Lb. Feldspar | gives | $120\times0.70=84.0$ Lb. Silica ($SiO_2$) |
| | | $120\times0.17=20.4$ Lb. Alumina ($Al_2O_3$) |
| | | $120\times0.07=8.4$ Lb. Soda ($Na_2O$) |
| | | $120\times0.06=7.2$ Lb. Potash ($K_2O$) |
| 72 Lb. Quartz | gives | $72\times1.00=72.0$ Lb. Silica ($SiO_2$) |
| 80 Lb. Borax | gives | $80\times0.16=12.8$ Lb. Water ($H_2O$) (Vaporized) |
| | | $80\times0.37=29.6$ Lb. Boric Oxide ($B_2O_3$) |
| | | $80\times0.47=37.6$ Lb. Soda ($Na_2O$) |
| 30 Lb. Cryolite | gives | $30\times0.44=13.2$ Lb. Soda ($Na_2O$) |
| | | $30\times0.24=7.2$ Lb. Alumina ($Al_2O_3$) |
| | | $30\times0.54=16.2$ Lb. Fluorine ($F_2$) (Vaporized) |

| | | |
|---|---|---|
| 7 Lb. Saltpeter | gives $7 \times 0.47 =$ | 3.3 Lb. Potash ($K_2O$) |
| | $7 \times 0.53 =$ | 3.7 Lb. Nitrogen Oxide ($N_2O_5$) (Vaporized) |
| 7½ Lb. Cobalt Oxide | gives $7\frac{1}{2} \times 1.00 = 7\frac{1}{2}$ | Lb. Cobalt Oxide (CoO) |
| 1 Lb. Manganese Oxide | gives $1 \times 1.00 = 1$ | Lb. Manganese Oxide ($MnO_2$) |

We shall consider that of the above the $H_2O$, $N_2O_5$ and $F_2$ are vaporized. This leaves for the constituents of the frit (totals of the above)

| | Lb. | Per Cent |
|---|---|---|
| Silica ($SiO_2$) | 156.0 | 53.5 |
| Alumina ($Al_2O_3$) | 27.6 | 9.5 |
| Soda ($Na_2O$) | 59.2 | 20.3 |
| Potash ($K_2O$) | 10.5 | 3.6 |
| Boric Oxide ($B_2O_3$) | 29.6 | 10.2 |
| Cobalt Oxide (CoO) | 7.5 | 2.6 |
| Manganese Oxide ($MnO_2$) | 1.0 | 0.3 |
| Total | 291.4 | |

The loss, theoretically, on smelting is plainly 317.5 — 291.4 = 26.1 lb. or $\frac{26.1}{317.5} = 8.2$ per cent. (317.5 lb. = original weight of batch).

28 The mill additions together with the agents used in "setting up" are fused with the frit as the piece is fired. By a process similar to the above, we could compute the final composition of the enamel by taking these additions into consideration. The actual amount of material added in this case, however, is so slight and of such a nature that the change resulting therefrom is not sufficient to affect materially the composition of the enamel. When an addition of about 12 per cent tin oxide accompanies the clay, a very significant change in the composition of the enamel is produced.

29 It may be well to note in passing some of the means by which various colors are produced in the enameling industries. It will be impossible to enter into great detail without taking too much time, but the mention of certain compounds in connection with the colors produced by their use will serve our purpose.

30 The production of a good white enamel either for cast-iron or sheet-steel work may be said to depend, at the present time, upon the use of tin oxide. Great have been the efforts to substitute less expensive substances, such as compounds of antimony and lead. But an antimony white which looks good alone is plainly seen to be off-color when compared with a good tin oxide white.

31 Going to the other extreme of color, black, we encounter difficulties. There are a large number of formulae for black enamels, but when the results are closely compared we find that the colors range widely through brown blacks, blue blacks, purple blacks, etc. Certain compounds of manganese and iron used together give a color approaching black. Other formulae call for the combined use of oxides of manganese, cobalt and copper. Again we find oxide of nickel added to the above three oxides.

32 A color much seen in enamels is blue and the use of cobalt is very satisfactory in the production of this color in various grades of intensity. Manganese alone produces purples and violets and in combination with cobalt gives various shades of purple-blue.

33 Green enamels are chiefly produced by the use of chromium oxide and copper oxide, while in some cases a mixture of copper and cobalt oxide is used.

34 Reds of various shades are produced by the use of red oxide of iron. In connection with it we find that tin oxide aids greatly in giving opacity and bringing out the color. In the production of brown enamels we may use ferrous chromate. Various yellows are produced by salts of cadmium, chromium and uranium.

35 The more delicate shades of rose and purple are produced by the use of gold compounds. So-called "pink-rose" is used in the manufacture of certain artistic enamels. Perhaps the best known gold compound used in enamel coloring is "purple of Casius." The exact composition of this product is a question. It is made by the combined use of auric, stannous, and stannic chlorides. The color produced is also commonly called "purple of Casius."

36 Before drawing this paper to a close, attention is invited to a general consideration of the future of the enameling industry. Neglecting art enameling and sign making we come to the field of steel enamels. So far as the cooking-ware industry is concerned, the field is practically constant. Granting that the demand for that class of article is increasing, as the public becomes accustomed and educated to its use, there is an opposing tendency in the rapidly increasing use of aluminum ware. Exactly how these and other factors now balance would be difficult to ascertain. But aluminum is a metal and its metallic properties cannot be denied. Under certain conditions it is attacked by various substances used in the culinary arts and a contamination of the preparation is inevitable. No doubt the time will come when a high silica enamel known to be free from tin and

other poisonous compounds will enter the cooking-ware field. The government is becoming more and more careful in protecting the public from foods of injurious nature and it is not too much to expect that soon it will establish more rigid restrictions relative to the ingredients entering into the manufacture of apparatus in which food is to be prepared. At such a time an enamel coming up to requirement will be free from injurious compounds and will come into great demand.

37 In the preparation of foods on a factory scale, we find an enormous and constantly increasing demand for larger pieces of enameled steel apparatus in the form of pans, kettles, tanks, pipe, etc. There are many lines of pressure being brought to bear both by the government and public opinion which lead to the conclusion that the increasing demand for this style of apparatus is without limit. Canning and preserving factories and dairy establishments have found a large use for copper and tin in the construction of containers, vacuum pans, etc. The acids of fruit juices and vegetable pulps have a very marked action on these metals and the resulting contamination of the product is known to be of danger to the consumer. The use of an enamel containing tin, lead or other metallic oxides is but the first step in the right direction. The presence of these metallic oxides in the enamel renders it corrodible and contamination results. The solution is the use of an acid-proof enamel free from all such poisonous substances. In the milk industry a similar line of reasoning applies. Further compare the ease of maintaining sanitary conditions in a one-piece enamel-lined unit with the trouble experienced in the use of a metal container or even an enameled article made up of composite parts between which are gaskets.

38 Another consideration relates to the preparation of chemicals later used in food products, for instance, baking powder. Many operations connected with the manufacture of such products have been carried on in lead or other metallic pans and the resulting contamination has given no end of trouble. Acid-proof enamel is rapidly solving this problem also.

39 Finally consider the chemical manufacturing processes now carried on in apparatus of lead, wood and earthenware, necessitated by the corrosive actions of the liquors and gases involved. This includes the pharmaceutical field which alone is a matter of great importance. In all these and many other lines the use of acid-proof enameled apparatus is rapidly finding and filling a great demand.

40 Not only does steel apparatus meet the demands of modern industries, but in case a cheaper product is desired and at the same time a heavier construction is permissible, acid-proof cast-iron apparatus has its field. The possibility for size and variety of construction is, of course, more limited than in the case of sheet-steel apparatus.

41 In view of these considerations and many others which these have called to mind, we cannot but conclude that the use of enameled apparatus has just begun and with this extension of the long known art of metal enameling, a field of great industrial possibilities both for manufacturer and user has been opened. We may not be criticised as being over optimistic when we predict that in their ultimate stage of development the enamel industries will be ranked among the greatest of commercial activities. At such a degree of development the enameling industry will in no way deserve classification among the lost arts.

# No. 1415

# GEARS FOR MACHINE-TOOL DRIVES

By John Parker, Providence, R. I.

Member of the Society

This basis of this paper is the consideration of the following six questions relating to the use of gears for driving machine tools:

1 Under what conditions is it advisable to use cast-iron or steel gears for machine-tool drives?

2 Are the objections to cast-iron on the ground of wear or breakage?

3 What tooth pressure is safe for cast-iron gears?

4 What grades of steel give best results and how should they be treated?

5 How hard is it advisable to make steel gears before machining them?

6 Are they to be hardened after machining, and if so, to what scleroscope test?

2 *Conditions under which it is Advisable to use Cast-Iron or Steel Gears for Machine-Tool Drives.* There are a number of well established gear conditions that are common to the majority of machine tools, which if noted may prove some guide to the selection of the proper material for the gears, considered from the standpoints of economy, efficiency, and durability. The conditions may be classified, as in Table 1.

3 *The Objections to Cast Iron.* The objections to cast iron cover both wear and breakage. If the speed is excessive, say above 500 ft. per minute, they are likely to wear quite rapidly; and on slow speeds and heavy pressure breakage will occur, unless they can be made of adequate size, as in the case *E,* where the back gears are so located in the machine that it is possible to employ large diameters, coarse pitches, and wide faces.

4 *The Safe Tooth Pressure for Cast-Iron Gears.* The question of tooth pressures in cast-iron gears is somewhat problematical. The

---

Presented at the Annual Meeting, 1913, of The American Society of Mechanical Engineers.

TABLE 1 GEAR CONDITIONS COMMON IN MACHINE TOOLS

| | | | Material |
|---|---|---|---|
| *A* | Gears always in mesh, the wear on the teeth being constant | (*a*) Slow speeds, light duty<br>(*b*) Slow speeds, heavy duty<br>(*c*) Fast speeds, light duty<br>(*d*) Fast speeds, heavy duty | Cast iron<br>Machinery steel<br>Machinery steel<br>Machinery steel, case-hardened |
| *B* | Gears in sets that are removable and interchangeable with each other, distributing the wear over a number of gears | These are change gears used in thread cutting on lathes, spiral cutting on milling machines, indexing on automatic gear cutters and feed and speed change gears; speeds and pressures are generally moderate | Cast iron, excepting the smallest, which may require to be of steel |
| *C* | Gears in sets that are non-removable and partially interchangeable, distributing the wear over a number of gears. Changes made while gears are in motion[1] | Used as quick-change feed gears—changes made by levers; speeds and pressures moderate | Machinery steel, case-hardened |
| *D* | Gears in sets that are non-removable and partially interchangeable, distributing the wear over a number of gears. Changes made when gears are at rest[2] | Used as quick-change speed gears—changes made by levers; high speeds and heavy pressure | Machinery steel, case-hardened |
| *E* | Gears that are employed only part of the time the machine is working, and are engaged and disengaged when the machine is stopped | This condition applies to back gears for the spindle drive. Gears are made large diameter, coarse pitch and wide face; speeds moderate and heavy pressure | Hard, close-grained cast iron |

[1] If the changes were made when the machine was at rest, the gears would not require hardening. But custom demands that changes be made while the machine is running.

[2] Although the changes are supposed to be made when gears are at rest, careless workmen will violate this rule, with the possibility of breaking the engaging gears. Some makers use an alloy steel in their spindle train to prevent breakage, but a better way is to provide means whereby it is necessary to stop the machine before throwing in the gears. This applies to the tumbler type of change gearing.

Brown & Sharpe Manufacturing Company have in successful operation a gear in the spindle drive of their largest milling machine made from a hard, close-grained cast iron having a tensile strength of 23,000 lb. per sq. in., which when running at the slowest speed sustains a pressure on the teeth of 8250 lb. It is calculated that two teeth are always in contact, which gives 4125 lb. pressure per tooth. This gear has 2½-in. face, 50 teeth of 3 diametral pitch and meshes with a pinion of 19 teeth. The area in cross-section of each tooth is 1¼ sq. in., equaling 3300 lb. per sq. in.; when the gear runs at the fastest speed the pressure is about 1000 lb. per sq. in. It is not known whether the pressure could be increased to any considerable extent, but it has been overloaded to at least 30 per cent without injuring it; this was when testing out the machine and the overload was of short duration. It might be said that this gear is not subjected to any sudden shock; if it were, the allowable tooth pressure would be considerably less.

5 *The grades of Steel that have given Best Results and how They have been Treated.* For gears that are of small proportions and yet are subjected to heavy duty, it has been found that in cases where the more common steels have failed, excellent results have been obtained from using a 5 per cent nickel steel. This steel case-hardens with a very hard surface and still has a strong and tough core, making it an ideal steel to use where the pressure is heavy or the gear is subjected to shock. Experience shows that drop forgings are more uniform in texture than bar stock. This grade of steel is given an oil treatment and is also annealed before machining; the oil treatment is as follows: heat to 1550 deg. fahr. and quench in oil. To anneal, reheat to 1350 deg. fahr. and cool very slowly. It is then ready to machine.

6 After machining, it is carbonized as follows: pack in any good carbonizing material and cover very carefully to exclude air, place in furnace and heat to 1700 deg. fahr., and hold long enough to get the desired depth of casing. Care should be taken to have it heated entirely through. Ordinarily three to four hours will suffice for this process. Then take out of furnace and cool off in the boxes; remove from the boxes and place in furnace or bath; reheat to 1550 deg. fahr. and quench in oil. Again reheat to about 1380 deg. fahr. and quench in oil or water according to the size and shape of gear. If the gear is of generous dimensions and free from sharp corners, water is preferably used. Small slender gears are quenched in oil, on account of the liability of cracking if water is used. For ordinary

gears the scleroscope test should show 80 to 85 points of hardness. If the gears are used as clash gears they should be drawn to 475 deg. fahr., or about 70 to 75 points of hardness, by scleroscope test, to avoid chipping.

7 *Degree of Hardness Advisable for Steel Gears before Machining Them.* The various kinds of steels used for gears are of such a nature that they do not call for treating before machining, but where extra toughness in shafts is required to withstand torsion and bending strains, 3½ per cent nickel steel is very satisfactory. This grade of steel is rough machined, then heat treated, as follows: place in open furnace or bath, heat to 1500 deg. fahr., and quench in oil. It is advisable to experiment with a small quantity in each batch before subjecting a whole lot to the drawing out heat, which should commence at about 700 deg. fahr. If the scleroscope registers between 50 and 58, the correct hardness has been obtained; if higher than 58, the parts should be reheated to a higher temperature than before; if lower than 50, the parts must be rehardened. After this treatment, the pieces are finish machined. No further hardening is necessary. When machining, slow speeds and feeds must be used.

8 *Hardening after Machining, and the Scleroscope Test.* Practically all alloy steels and all low-carbon steels are hardened after machining and finished by grinding after hardening. About 0.010 in. on the diameter is left for this operation. All gears should run true, and to obtain this result not only are the holes ground true with the pitch circle, but the hubs are ground on their faces so they will set square with their shafts when tightened up by nuts. The scleroscope test for 30 to 35 point carbon machinery steel is anywhere from 80 to 90, and 5 per cent nickel steel for ordinary gears 80 to 85, and for clash gears 70 to 75. All steels are tested by the file in addition to the scleroscope. The file test by an expert is very reliable and some feel that possibly more confidence can be placed on his judgment than on any testing instrument.

9 The above notes apply to spur and bevel gears. For worm and worm-wheel drives, the worm should be made of machinery steel, case-hardened, and the wheel of a hard bronze. Both should run in a bath of oil, especially if under high speed and heavy duty. Spiral gears should be used only where the duty is light. The material should be the same as for a worm and wheel, and they should also run in oil to avoid cutting.

10 For index mechanisms, where accuracy is essential, if the

worm is hardened the thread must be ground afterwards. This is done in all the spiral threads of Brown & Sharpe's make. Generally, the worm, made of tool steel, is left soft. Worm wheels used for indexing purposes only are usually made of cast iron, and invariably if of large diameter. High-multiple threaded worms for indexing mechanisms should not be used; a double thread can be tolerated, but not more, if accurate indexings are required.

## APPENDIX

A letter embodying the six questions set forth in Par. 1 of the paper was sent out by the Committee on Machine Shop Practice to various machine builders, and the following replies were received:

J. B. Doan.[1] Nearly all the gears used today in machine-tool construction are of cast iron, although the quantity of steel gears is increasing. The American Tool Works Company use cast iron with slow peripheral speeds and comparatively large diameters; steel for small diameters and high speeds, hardening those which experience teaches need such treatmnet.

The main objection to cast-iron gears is their breakage, and not so much their wear. A good grade of cast iron shows very good wear. A great many steel pinions are used, some of them hardened, working with cast-iron gears of large diameter, the difference of material compensating in strength for the weakness of the pinion tooth, and also for the wear occasioned by the different number of teeth.

The Lewis formula is used by The American Tool Works Company for tooth pressures on cast-iron gears.

For hardened-steel gears the following method with special gear stock made of very low carbon is employed: The gear is machine finished with an allowance of 0.012 in. on the bore for grinding after being heat treated as follows:

*a* Pack in round cast-iron box, using 10 per cent charred leather, 40 per cent burnt bone, 50 per cent raw bone
*b* Seal top with mortar of iron filings and fire clay about 1 in. thick
*c* Cover with cast-iron lid and lute with clay
*d* Heat for nine hours to 1560 deg. fahr.
*e* Remove box from furnace and cool without disturbing contents
*f* Remove pieces and heat to 1550 deg. fahr. in furnace
*g* Quench in fish oil
*h* Draw in tempering oil to 475 deg. fahr.

Gears are hardened after machining and tested with the scleroscope to 80.

J. B. Green. Gearing for machine tools is one of the interesting problems in connection with their design and construction. The selection of suitable materials for gears is usually fixed by the character of the machine and its uses. Some machines permit the use of large dimensions while in others the sizes are often very limited.

The Lewis formula with the modifications made by Carl G. Barth as given

[1] Vice-President and General Manager, The American Tool Works Company, Cincinnati, Ohio.

in the American Machinist Gear Book, Pages 57 and 58, is the well beaten path usually followed in selecting the best materials for gearing, and when properly used, there is a reasonable field of uses for all materials.

Cast-iron can be used in all places where there is sufficient room for the proper dimensions and, when not overloaded, the teeth wear to a polished and glazed surface so characteristic of this material. When slightly overloaded, the wearing surface of the teeth becomes crushed and cut at the pitch line and then wears out very rapidly; when considerably overloaded, failure occurs through breakage.

Steel iron, or semi-steel, as it is sometimes called, is made of a mixture of steel and cast iron and when made of the proper proportions, is a very good improvement over cast iron, being strong enough to carry about 50 per cent more load and giving very similar results in regard to wear and breakage.

Low-carbon steel gears will carry from two to two and one-half times the load for cast iron; when slightly overloaded they wear a line depression across the face of the teeth at the pitch diameter and when continuously overloaded soon wear out.

Steel gears containing about 20 per cent carbon can be case-hardened after being machined; when so treated they will stand considerable use without wear and are especially valuable for pinions meshing into large gears.

The high-carbon and alloy-steel gears will stand loads proportionate with the permissible stresses for the various materials. These steels may be heat-treated before machining which will allow large pieces to be greatly toughened and strengthened and then finished to the required accuracy, whereas, if the same pieces were hardened after machining, the resulting distortion would make such an operation impossible.

When the nature of the piece will permit, the alloy steels may be hardened throughout when all machining is completed and then drawn to the proper temper so as to make an almost indestructible gear. The exact hardness or toughness of teeth must be determined by the uses for which the gear is intended and in many cases experiments are necessary.

E. A. Muller. If cast-iron gears were inadequate for their purpose, I should be the first to discard their use. In view, however, of the most excellent results obtained with them, such a move would not be warranted, especially when the substitution of heat-treated alloy-steel gears would simply increase the cost of a machine.

If a large amount of mechanism subjected to heavy loads must be placed in a small compass the cast-iron gear is unequal to the requirements, but in many machine tools much space is available for the driving gears, and there is really no excuse for adding to the cost of these machines by designing the equivalent of automobile transmission gears for machine-tool drives.

Since 1893 I have used Wilfred Lewis' formula for safe working pressure on gear teeth with uniformly excellent results. Neither wear nor breakage has been objectionable; in fact, gears in use six years on machines in hard service show no appreciable wear. Cast-iron gears and steel pinions proportioned in accordance with the Lewis formula should not exceed 1200 ft. pitch line velocity. Experience with gears designed by this formula indicates that cast-iron gears can transmit greater horsepower without failure, but that wear begins to increase considerably as the load increases.

The King Machine Tool Company do not use heat-treated or hardened-steel gears at all. In such machinery where ample space is provided for gearing, I do not believe that properly proportioned cast-iron gears and steel pinions can be improved upon. However, where conditions obtain as in automobile transmissions, heat-treated and hardened-steel gears are indispensable.

Another machine builder writes: We find that cast-iron and mild-steel gears work very satisfactorily for machine-tool drives wherever there is room enough to make them of the proper size to avoid extreme wear. This is particularly true where the gears can run in oil. In general, of two sets of gears of the same dimensions, running under the same load, at the same speed, the one made of unhardened steel will run more smoothly than the one made of hardened steel. The question of hardened versus unhardened gears is therefore one of space and weight limitations. It should be noted that cast iron running with cast iron, or a steel pinion running with a cast-iron gear is a good combination, but that two mild steel gears running together, particularly at high speeds, should be avoided under any circumstances. The soft-steel gears abrade themselves with great rapidity.

The objections to cast iron are on the ground of liability to breakage in the case of work involving hammer blows or sudden or reversed stresses. There is no great objection from the standpoint of wear, where gears of the proper size can be used as indicated in the preceding paragraph, and where constant lubrication is provided.

For ordinary machine tool work we have found by experience that the Lewis formula, with the strength factors originally given, gives gears that are from 25 per cent to 40 per cent too strong.

Harder steel gives better results from the standpoint of wear and abrasion as well as of strength. When it becomes necessary to use hardened gears, there is nothing better than to follow automobile practice closely, using chrome nickel or similar materials heat treated in the approved manner.

We should be inclined to anneal them as thoroughly as possible before machining in order to have the tooth surfaces cut as accurately as possible. At the same time, with the shaper tool of the gear shaper it is possible to machine gears for a finishing cut that are heat treated so hard that a file will barely touch them. This is done by several firms, using turpentine as a lubricant, with very good results.

We have found that the scleroscope test is unreliable, and that its indications show very little regarding the matter of hardness for resisting wear, particularly when trying to compare pieces of different shapes or methods of support. Some gear specialists have come to this conclusion also, while others use the instrument with more or less favorable results. We would appreciate further information on this point.

## DISCUSSION

John Riddell referred to the necessity for correct design in gears and to the attention which they should receive in the process of manufacture in pattern shop, foundry and machine shop.

If the foundry is large so that it can set apart a cupola or a melt in a certain cupola, the results may be all that can be expected from cast gears, but if the foundry is of medium size and makes no particular melt for gears, common cast iron that may be very soft and porous is the result. Cast iron for machine tools and other apparatus generally will vary somewhat with the foundry and even the section of country in which such castings are made.

The writer believes that he was among the first in the United States to use the hobbing process for cutting gears. As originally started there were some drawbacks with the process and there is still something to learn.

Various attempts have been made to improve the indexing device, some of which appear to be very good, but others of which, while eliminating one error, have introduced several other irregularities, and do not seem to have made the general improvement expected by the designer.

As to the other materials outside of cast iron and gun iron, great improvements have been made, due to the automobile and high-speed requirements, such as in turbine engines and the connection of electric motors with machine tools. The latter has been made by the use of cloth pinions.

These pinions seem to have a number of advantages over other types of non-metallic pinions and have proved in a wide variety of applications to have strength about equal to that of cast iron. The Lewis cast-iron formula is used in their design and many users believe that by virtue of their elasticity they withstand service shocks better than cast-iron pinions of the same dimensions. Their life in most applications appears to be a little longer, and their substitution for cast iron on a very large scale is, therefore, only a question of whether their higher cost is justified by their more quiet operation.

The gear may be made of machinery steel or a casting, but the pinion is usually of harder material and very frequently oil tempered, unless cloth or rawhide is used.

Gear boxes should be used in some places but not in others. Sometimes when in a hurry for machine tools, gear boxes are ordered for the initial drive, where ordinarily the writer would prefer to use a variable-speed motor to take the place of unnecessary gear boxes. No matter how well these gears are made they are bound to make too much noise. A gear box for changing feeds and for automobiles, perhaps is necessary at present.

Much is yet to be done in designing machine tools as to gears for driving the machine as well as for giving different feed variations. On feeds of large machine tools, such as planers, and large planer type milling machines, electric motors are being very successfully used, which not only cut out a lot of gears but other mechanisms, and are a step in the right direction. Gears, especially bevel gears, have always seemed a necessary evil, though many attempts have been made to use spiral and helical gears to eliminate some of the difficulties inherent in bevel gears.

The writer would like to urge upon the trade, a more accurate system both of indexing and of mounting gears after they have been properly indexed and cut. If a gear is allowed to have the slightest play on a shaft it will run eccentrically and no matter how little this eccentricity may be there will be a noise which cannot possibly be eliminated.

F. V. McMullin (written). Having had experience in furnishing forgings to gear makers, I can say that today, their specifications are based mostly on chemical composition of the steel, some on physical tests, while a very few give both chemical and physical. For the ordinary gear that is neither case-hardened nor heat treated, gear makers usually call for 25 to 30 point carbon, no other element being mentioned. If the gear is to have hard usage, they will call for a higher carbon, 40 to 50 point, or even 60 point being demanded; some also specify the manganese, sulphur and phosphorous.

Gears that are to be case-hardened should be made of steel of not more than 20-point carbon, as this more easily takes the case. Some makers still buy 30-point carbon steel for case-hardening, which in my opinion is unwise. When gears are to be oil tempered a 40 to 50-point carbon steel or a 3½ per cent nickel 40-point carbon steel should give satisfaction. A 3½ per cent nickel, 25-point carbon steel is used both with and without heat treating, though I think the latter is not good practice, as such steels should always be treated. Where a gear is to have hard wear and the design permits the use of a thick tooth, high manganese steel will be found satisfactory. One well-known machine tool builder uses a high manganese gear in mesh with a train of heat-treated nickel chrome gears with very good results.

I append a few typical gear specifications from our order files:

*a* Carbon 20 to 30 point; Phosphorus and sulphur less than 0.04

*b* Carbon 0.07 to 0.10; Manganese 0.30 to 0.40; Sulphur 0.035; Phosphorus 0.01, and Silicon 0.01

*c* Carbon 0.40 to 0.50; Manganese 0.50 to 0.55; Phosphorus 0.01 to 0.015, and Sulphur 0.035 to 0.04

*d* Carbon Steel Annealed: Ultimate tensile strength 80,000 lb.; elastic limit 40,000 lb.; elongation 22 per cent in 2 in.; contraction of area 35 per cent

*e* Nickel Steel, or Nickel Chrome, or Carbon Steel: Phosphorus and sulphur less than 0.04; ultimate tensile strength 80,000 lb.; elastic limit 50,000 lb.; elongation 25 per cent in 2 in.

*f* Carbon Steel, 55 to 65-Point Carbon: Ultimate tensile strength 85,000 to 100,000 lb.; elastic limit 45 per cent of ultimate; elongation 22 per cent in 2 in.; contraction of area 35 per cent

A. L. De Leeuw (written). In regard to the use of hardened clash gears, we replaced all soft gears by hardened ones, in our milling machines, not because the soft gears were not satisfactory, but because it was impossible to control the operators handling the milling machines. If the operator would follow the injunction not to throw in the gears when the machine was running, soft gears would have been entirely satisfactory.

One of the speakers said that in general the trade demanded steel gears. This is a clear case of the minority ruling. In 99 out of 100 cases, cast iron is good enough; but in the hundredth case there is trouble; this trouble is noised about, and then the conditions of competition make it absolutely necessary to change over to steel, even though cast-iron gears might have a number of advantages of their own.

In regard to the use of cast-iron gears for slow drives, it has been the experience of a good many engineers, and, to a certain extent, has been my own experience, that the Lewis formula is too conservative for low speeds. Pressures may be used, 50 per cent, and in some cases even 100 per cent, above the figures of the Lewis formula.

I want to call attention to Table 1, in which the author speaks of gears in sets that are non-removable. He mentions that each one of these gears is used at times, thus distributing the wear over a number of gears. This idea may be in the mind of the designer, but whether all of these gears are actually going to be used is a matter over which there is no control after the machine enters the

shop of the customer. It may be that a single set of gears will be used all the time.

H. F. L. Orcutt said that in Great Britain the gear box drive is coming in, copying the American practice, and it is recognized that the gear box drive and the single pulley drive are the developments of the future.

The requirements of the London police on the English buses may be of interest. The makers have had to use chain drives in the transmission box to satisfy the demands of the police for quiet transmission. He thought those who had been in London and witnessed the running of the buses could never have heard more quiet transmission.

A. W. Thompson said that he was primarily interested in the manufacture of gears for textile machines and spoke of the satisfaction he had found in the hobbing process for cutting spur gears both as regards economy of manufacture and smoothness of running In textile work the use of gears with teeth cut at an angle is favored for special purposes where it is desirable to eliminate vibration, for which the hobbing process is particularly well adapted.

E. H. Neff said he considered the transition from cast-iron gears to steel ones as an inevitable step in the development of machine tools due to the demand that such tools be capable of continuous use. The user of machine tools has been interested in gear failures only as they effect the continuous operation of his factory equipment. The machine tool builder has been interested in gear failures from the standpoint of the adequacy of his designs. If a cast-iron gear broke occasionally due to defective casting or unusual shock, it must be so made in future that such breakages will not be duplicated. The only option, therefore, was to increase the pitch of his cast-iron gears, which would probably make them clumsy, or change the material to steel, by which a very material increase in strength and reliability could be obtained without changing the design of any of the parts. The gradual crowding out of cast-iron gears by steel gears is, therefore, a case of the survival of the fittest.

F. DeR. Furman (written). There seems to me to be a pretty close link between item *A* and the different materials that are there given in the third column of Table 1. The author uses the words, "Gears always in mesh, the wear on the teeth being constant." I doubt whether this is a proper expression, because the wear on the

teeth, as I see it, is greater at the tips, there being more sliding as the line of contact passes away from the line of centers; and if there is more sliding at the ends of the teeth, naturally there should be more rapid wear at that place. And then, as there is more wear, it leaves the bulk of the work to be done by the teeth on or near the pitch line. Now, if we have a material that is soft enough to wear away fast, under that rubbing action, the pressure between the teeth will take place near the centers of the teeth and the work of transmission will be better done. And if the tooth is made of a hard material which refuses to wear away or to yield readily as the rubbing action takes place at the ends of the teeth, that hard tooth is going to receive more punishment at the ends and more stress in the teeth will be set up than in the tooth made of softer or more yielding material. The center of pressure which produces the greatest stresses in the teeth, travels from the tip of the tooth to the center and back again alternating until the teeth are worn out or until they break, but in the soft or yielding material the most injurious pressure which is at the tip of the tooth will last for the shortest period of time. Of course, on the other hand, the tooth that is made of hard material is stronger to receive the punishment, but I think in considering the materials that are used, we can get along better if we also consider the action which occurs between the teeth.

C. R. Gabriel mentioned a case of high-speed gearing in connection with automobile torpedo engine construction showing the possibilities of gear drives at high speed and heavy duty. In the torpedo an engine of the turbine type is used to drive the propeller shafts through a balanced train of eight gears, made of chrome nickel steel, case-hardened, the teeth of the spur gears being ground after hardening These gears work under very heavy duty, considering the proportions of the gears.

The engine develops 125 h.p., but due to the fact that the turbine motors, which run in opposite directions, do not develop equal amounts of power, one set of gears is required to transmit 75 h.p. The spur gears consist of pinions with 24 teeth and gears of 80 teeth, 10 pitch and 1⅛ in.-face, making a ratio of $3\,{}^1/_3$; the bevel gears have a ratio of three to one, the pinions being 1½-in. pitch diameter and the mating gears 4½ in., also 10 pitch. The speed of these gears is very high, that of the spur gears being 7500 ft. per min., and of the bevels 1400 ft. per min., the tooth pressure on the bevels being about 1800 lb.

This construction has been well tried out, several hundred torpedos being fitted with engines of this type. While the length of service is not such as gears in machine tools and automobiles are required to stand, some of these torpedos have made several hundred runs of 8 min. to 10 min. duration, and gears examined after this amount of service showed no appreciable wear. It should be mentioned that these gears are never subjected to shock, as the engine starts with an accelerated motion, and the load is practically constant throughout the run. In operation these engines are very noisy, but in torpedos this is not an objection, as when submerged in the water the noise is not heard.

The Author. In regard to Mr. Riddell's statement, that he did not like the variable speed mechanism in his machines on account of the noise and the use of gears in general, I believe the spindle speed gear box to be one of the best things we ever put in milling machines. We find that we can do about four times the work in the milling machines by the constant speed belt drive and variable speed mechanism, than with the old cone pulley belt drive. As to the noise, we have no trouble in the milling machines made by the Brown & Sharpe Manufacturing Company; the hardened gears run very nicely and quietly.

Regarding Professor Furman's criticism of the gear conditions specified in class *A*, what I meant as to the wear of the teeth being constant was that the teeth are always engaged, and there is no moving from one gear to another, such as the gears in the milling machine knee, where they are always in action when work is being done.

Mr. De Leeuw's statement that the hundredth case might be using only one speed is very true; but I have treated this subject in a general way. Of course, we have different conditions to meet and the machines are designed for the general run of the trade.

## No. 1416

# CAST IRON FOR MACHINE-TOOL PARTS

By Henry M. Wood, Cincinnati, Ohio

Associate Member of the Society

When considering what sort of iron to use for various purposes, the machine-tool manufacturer is interested chiefly in the result attained by the foundryman, i.e., the strength, soundness, hardness, etc., of the casting, rather than the foundry methods used in accomplishing such a result. Nevertheless as an investigation of the character of the casting is made through chemical analysis, it is well first to consider the effect of each of the elements usually present in cast iron.

### INFLUENCE OF CHEMICAL ELEMENTS

2 The five elements present in ordinary cast iron are carbon (both free as graphite and combined), silicon, sulphur, manganese, and phosphorus.

3 *Carbon.* Carbon exerts a more important and direct influence on the quality of the metal than does any other element. The percentage of carbon in the casting and the form in which it exists are dependent upon the melting conditions, the thermal treatment, and the amounts of other elements present. It is therefore necessary to take account of the effects of the carbon itself and the way its influence is modified by the presence of varying quantities of other elements. To obtain a proper relation between combined and free carbon is the most important point. The percentage of total carbon usually ranges from 3.5 to 4.

4 *Graphitic Carbon.* Graphite is merely mixed with the iron instead of being in chemical combination. Since it is only mixed with the metal it cannot exert any direct influence upon the properties of the molecules of the iron. So far as the graphite itself is concerned the toughness, hardness, and melting point of the grains of the iron will not be altered. The tensile strength will, however, be greatly affected, since the interposition of the flakes of graphite will act as

---

Presented at the Annual Meeting, 1913, of The American Society of Mechanical Engineers.

partings between the grains of the metal and reduce the cohesion. Iron high in graphite is soft and can be readily machined, but is of low tensile strength.

5 *Combined Carbon.* The carbon which is in chemical combination affects directly and greatly the properties of ordinary cast iron. It is the principal factor in determining the hardness, tenacity, soundness, and freedom from internal stresses of the castings. In general the percentage of combined carbon ranges from 0.05 in the softest cast iron to about 0.60 in iron of the highest elasticity. With suitable iron mixtures the amount of silicon and sulphur present regulates separation of carbon as graphite so that the amount of silicon present is an index of the relation between the free and combined carbon. Many analyses submitted do not state the percentage of carbon, perhaps because the foundrymen do not appreciate the importance of it, or perhaps because they consider the quantities of silicon, manganese and sulphur, which influence the quantity of combined carbon, make it unnecessary to determine the latter.

6 *Silicon.* Silicon tends to cause separation of the carbon as graphite. As the hardening effect of the silicon necessary to do this is less than that of the combined carbon converted into free carbon, the net result of adding silicon is softer iron. Under conditions which are easily regulated by selection of materials and in other convenient ways, the general influence of silicon can be utilized to render the cast iron suitable to various purposes, but it must be used with due regard to the other constituents of the metal.

7 *Sulphur.* The influence of sulphur is opposite to silicon, in that it makes the iron harder and more brittle. It makes the iron more sluggish when pouring and is liable to cause unsound castings. Sulphur increases shrinkage, resulting in internal stresses of unknown value and sometimes producing distorted castings. This element should be kept low, especially in small castings.

8 *Manganese.* One effect of manganese is to combine with sulphur and go off in the slag. Another is to retard the separation of graphite by combining with carbon, and in this its influence is opposite to that of silicon, rendering the iron harder, stronger and perhaps closer.

9 *Phosphorus.* Phosphorus lowers the melting point and increases the fluidity of the molten iron. On the other hand, phosphorus makes the casting harder, more crystalline and brittle, and reduces the tensile strength.

### SPECIAL MIXTURES

10 *Semi-Steel.* This mixture is obtained by charging mild steel scrap into the cupola. It is common practice to use about 20 per cent steel, but any amount up to 70 per cent may be used. The 20 per cent mixture usually gives the desired results for machine-tool work. Semi-steel is lower in total carbon than cast iron, seldom having more than 3 per cent. The fine grain of semi-steel is due to low percentage and fineness of graphite. It is used for parts requiring good appearance, tensile strength, and ability to resist shock. It is often used for large gear blank castings for various machine tools, for milling-machine tables, knees, and saddles, for lathe reverse plates, compound rests, etc. Tensile strength is relatively high and the grain of the casting is close.

11 *Alloys.* Elements sometimes introduced into cast iron for special purposes are vanadium, titanium, nickel and chromium. Vanadium increases the ability of the casting to resist wear. Titanium combines with nitrogen and goes off in the slag, thus acting as a scavenger and making closer and cleaner iron. While nickel and chromium are now used to a limited extent in heavy chilled castings such as car wheels and rolling-mill rolls, the writer has been unable to find instances of the use of these metals in cast iron for machine-tool parts or for similar purposes. Their use might be desirable for special castings as it would doubtless increase the strength. But for machine-tool work the additional expense of such alloy castings might not be warranted, as today all parts where high tensile strength is required are made of steel.

### MIXTURES USED BY REPRESENTATIVE MACHINE-TOOL MAKERS

12 In the belief that valuable information could be secured by a comparison of the present practices of representative machine-tool manufacturers, the writer asked a number of machine-tool makers in different lines and in different sections of the country if they would be willing to submit an outline of their practice. The following excerpts pertaining to the mixtures used and chemical analyses of the castings are quoted from their replies:

13 *A Builder of Special Machine Tools:* Our iron is bought on analysis specifications, covering two grades as follows:

| | |
|---|---|
| Two Plain | |
| Silicon | 1.75 to 2.25 |
| Manganese | 0.60 to 0.90 |
| Phosphorus | 0.50 to 0.80 |
| Sulphur | 0.05 and under |
| No. 3 | |
| Silicon | 1.00 to 1.50 |
| Manganese | 0.50 and over |
| Phosphorus | 0.50 to 0.80 |
| Sulphur | 0.07 and under |

In addition to the above we use materials as follows: No. 1 machinery scrap, mild steel scrap, manganese steel scrap. The last carries 12 per cent of manganese with quantities of the other elements so small that they are negligible in gray iron foundry work.

14 Our mixtures are figured out on the actual analysis of each car, insuring in the castings uniformity of analysis and consequently of physical characteristics, such as strength, density, and machining qualities. In general practice we use three different mixtures suited to our varying needs.

15 In our first mixture we include our lighter castings such as pulleys, small gears, washers, hand-wheels, brackets, and the like. In this mixture we endeavor to have the following analysis:

| | |
|---|---|
| Silicon | 1.90 |
| Manganese | 0.60 |
| Phosphorus | 0.70 |
| Sulphur | 0.08 |

This is usually secured by the use of a mixture of 50 per cent of two or three lots of plain iron and 50 per cent of scrap. The proportions of the different pig irons are adjusted to produce the proper analysis in the mixture, and the scrap is partly our own foundry return and the balance No. 1 machinery scrap.

16 Our second mixture covers all our heavy work, such as planer beds, posts, tables, face plates, frames, etc. These castings require strength and sufficient density to permit the machined surface to take a high polish. These ends we accomplish by an analysis as follows:

| | |
|---|---|
| Silicon | 1.40 |
| Manganese | 0.60 |
| Phosphorus | 0.60 |
| Sulphur | 0.09 |

This mixture consists of 45 per cent of two or more No. 3 irons and 55 per cent total scrap, shop and No. 1 machinery together. Should this mixture fail to yield sufficient manganese the addition of 1 to 2 per cent of manganese steel scrap is made to correct it.

17 The third mixture is semi-steel, used principally for large blank gears and castings requiring special strength. Its analysis is:

| | |
|---|---|
| Silicon | 1.20 |
| Manganese | 0.90 |
| Phosphorus | 0.45 |
| Sulphur | 0.09 |

Its average makeup is

| | Per Cent |
|---|---|
| Machinery scrap | 30 |
| Mild steel scrap | 20 |
| Manganese steel scrap | 5 |
| No. 3 pig iron | 45 |

18 The above mixtures cover our entire range of work except cases where some special composition is required or desired.

19 All materials are weighed before charging into the cupola and all due precautions are taken to insure proper melting conditions and perfect mixtures of the various materials entering into each charge.

20 *A Manufacturer of Precision Machinery:* In our work we run various grades of iron to meet the conditions existing in the machines or in the parts of machines under consideration.

21 In a general way our mixtures in per cent, run as follows:

| Silicon | Manganese | Phosphorus |
|---|---|---|
| 3.00 | 0.60 | 0.80 |
| 2.40 | 0.65 | 0.70 |
| 2.00 | 0.65 | 0.60 |

The first is for the average run of castings of smaller size; the second for the larger castings. Where we need a special close-grain iron we use the third mixture.

22 *A Manufacturer of Milling Machines:* We have never carried on any extensive experiments to learn the best mixtures of cast iron for our purposes. We use in the tables, knees, saddles and vises about 20 per cent of steel with a view to obtaining a close-grain casting, and increasing somewhat its strength.

23 We use practically no cast iron for gears or small parts, these being made of steel drop forged in the case of larger parts, and also in the case of smaller parts when not adopted for manufacture from the bar.

24 The subject of gray-iron castings is, we believe, one of the most annoying to be found in connection with the manufacture of machine tools. Customers are not satisfied to accept machines with defective castings even though the deficiency is of such a nature as to in no wise impair the life or efficiency of the machine.

25 The ideal casting is, of course, one that is so close as not to show any grain when finished and at the same time, just as hard as it can be, and be worked into shape.

26 The question of strength is probably not so important, as there is opportunity to use sufficient bulk to obtain strength. At any rate this is true of the parts that we make of cast iron, for, as stated above, all our gears and like parts are made from steel which is case-hardened.

27 *A Manufacturer of Heavy Lathes:* With the heavier castings we are using a semi-steel mixture with about 20 per cent of steel. The analysis of this iron shows 1.60 to 1.70 silicon, 0.65 to 0.75 manganese, 0.40 phosphorus, 0.8 to 0.10 sulphur. While our carbons are not noted as a rule, we get a check on these every once in a while, showing the total carbons about 3.50 to 3.60.

28 Our iron for smaller pieces runs from 1.80 to 1.90 in silicon, 0.40 to 0.50 in phosphorus, 0.65 to 0.70 manganese, 0.07 to 0.10 in sulphur. The total

carbon shows up practically the same in both mixtures. Our test bars on the first mixture break at from 2800 to 3200 and on the latter mixtures at about 2600. This refers to 1 in. by 1 in. standard bars supported on 12 in. centers.

29 *A Manufacturer of Grinding Machines:* We use castings with various proportions of steel according to the size of the casting and the place where it is to be used, so that today we have very bright lustrous surfaces and it is possible to get accurate alignment.

## PRACTICE WITH REFERENCE TO CHILLING CASTINGS

30 There is wide difference of opinion among machine-tool manufacturers as to the desirability of chilling any surfaces of castings. The writer asked some of the representative manufacturers of various classes of machine tools for their experience on this point. The following quotations from their replies state both sides of the case:

31 *A Manufacturer of Milling Machines:* We are not using any chills at this time, though we have experimented with these from time to time but have reached no satisfactory conclusion.

32 *A Lathe Manufacturer:* We have not used chills on any parts of our machines, which, we must concede, is from many points of view, not a very satisfactory admission to make.

33 *A Manufacturer of Heavy Machine Tools:* For quite a period, about nine or ten years ago, we chilled the ways on our lathe and also the rails on our boring mills. We found, however, after they had been out some time that there was quite a bit of trouble with the chilled surfaces scratching. It was hard for us to find exactly what was the root of the trouble and we finally gave it up. The effect of the scratching of the chilled ways was a most peculiar one, and we sometimes observed on a machine even before it had gone out that some little particle of material had settled on the way and scratched the same badly.

34 *A Manufacturer of Grinding Machines:* In regard to chilled iron, of course, you know that chilled iron means this and nothing else: It means iron that cannot be filed, planed or scraped. At least any mechanic who hears the words "chilled iron" understands it to mean just that thing, a surface that cannot be cut with tools. Now, of course, you realize that such a surface makes it impossible to get practical aligning ways on machine tools. It might just barely be possible to grind them accurately, but probably not practical to do so. We use an iron with steel mixture, and vary the mixture according to the size of the casting, and we produce a casting as dense and as hard as we can possibly plane and scrape with any surety of getting perfect alignment, because imperfect bearing and imperfect alignment is just as bad and just as sure an error as iron that would be too soft. In fact if one had iron that was exceedingly soft, and should choose to make ways which are very wide it might be more durable than one made with hard iron and smaller ways.

35 *Another Manufacturer of Grinding Machines:* The main reason for chilling the different parts of our work is to increase the wearing durability and at the same time get the advantage of refining the metal and a clean sur-

face. The parts chilled are the guides of the carriage and the surface of the table upon which the head and tailstocks are mounted.

36 Our method of chilling is to place plates of ⅝ in. in thickness in the mold, and these give a depth of chill of about ½ in. and a degree of hardness just to the point of where the machining can be readily done.

37 It would be natural to suppose that this chilling would produce or increase internal strains, but on our work, such conditions have not given any trouble.

38 Our work is not of such a nature where hammering upon it or peening is necessary. Therefore, we are unable to state just what action would take place as the result of hammer blows and peening.

39 The chilled surfaces are very much more durable than metal in the ordinary condition, and we believe by the chilling process the durability of the surface upon which wear comes is increased at least from 300 to 400 per cent.

40 *A Builder of Heavy Machine Tools:* In 1888 we began the practice of solidifying cast-iron surfaces by introducing chill blocks in the molds, and we have continued the practice ever since.

41 Answering your questions specifically, first, we have not discontinued the use of chilled surfaces because of any difficulty in oiling. We have never found that the fine grain of the chilled iron prevented the oil from sticking. There is no truth in the statement.

42 We have not found any increase in internal strains due to the use of chills. On the contrary, when chills are properly placed they equalize the cooling of the heavy parts adjacent to lighter portions and reduce the internal stresses which would naturally result from the difference in time of cooling. If the chills are improperly used it would be possible, especially in thin castings, to cool the entire mass too rapidly and produce internal stresses.

43 We have not found that the proper use of chills makes the iron more sensitive to a peening action; in fact, we have evidence to the contrary.

44 The chilled surface we believe to be more durable. We have cases where gearing made in this way outlasted several sets made in the old manner.

45 The success or failure of this process depends upon the ability to produce, day in and day out, the kind of metal required, and, further, the intelligent designing of chills of iron molds so that a proper relation may always be observed between the size and shape of the casting and the thickness of the mold or chill block.

46 A *Boring-Mill Manufacturer.* We are chilling certain surfaces on our boring-mill spindles with good success, but have found no occasion for chilling any other surfaces. If we experienced difficulty due to undue wear on sliding surfaces, we would increase the area of the surfaces and supply better lubrication and protection from dirt rather than to try to chill the surfaces of these parts.

47 The chills which we use on our spindles serve two purposes: first, by securing closer grained metal; second, by improving the quality of the wearing surfaces. We found that it was difficult to get good castings of these spindles until we did use chills.

48 *A Builder of Special Machine Tools:* Concerning chills, we would say that to some extent we are now using these on the surfaces of beds and similar castings.

49 *A Manufacturer of Precision Machinery:* We use such chills as may be necessary to give the surface in connection with the ways and moving parts.

CHILLED LATHE BEDS

50 *Analysis of Iron.* In view of the considerable differences in opinions of the value of chilled surfaces and the idea held by some that it is impossible to chill an iron of high tensile strength without

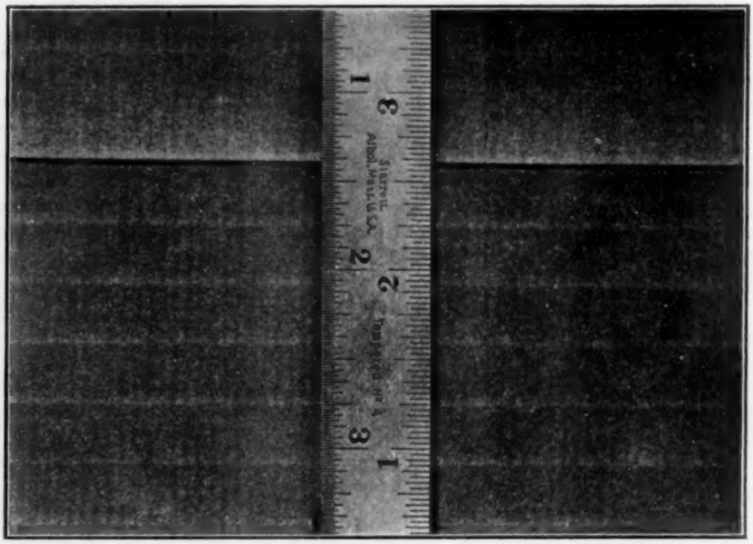


FIG. 1 CHILLED CASTING, FINISH PLANED BUT NOT SCRAPED

making it so hard it cannot be machined, the practice of The Lodge & Shipley Machine Tool Company, with which the writer is connected, is here outlined.

51 Three average analyses are as follows:

| Silicon | Sulphur | Phosphorus | Manganese | Tensile Strength |
|---|---|---|---|---|
| 2.16 | 0.065 | 1.01 | 0.40 | 22,310 |
| 2.17 | 0.065 | 1.01 | 0.39 | 24,840 |
| 2.45 | 0.076 | 0.63 | 0.71 | 24,195 |

The first analysis is of a specimen taken in January 1913 from the first iron run in a heat; the second, from the last iron of the same heat; the third, from the average iron of a heat in September 1913.

52 This same iron is used for lathe beds of which the ways are chilled, also for other cast-iron parts which do not require the high

tensile strength of semi-steel. For some parts, such as compound rest top slides and reverse plates, we use semi-steel. Parts subject to greater stress or to severe shock are made of steel.

53 *Appearance of Chilled Surface.* A portion of the finished way of the lathe bed as it comes off the planer and before scraping is shown in Fig. 1. A steel scale is laid on the surface to indicate the

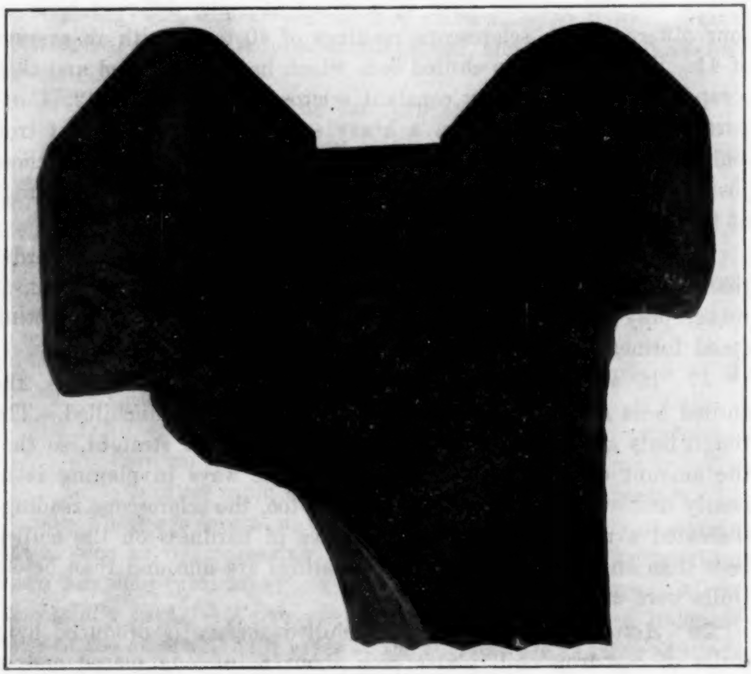


FIG. 2 CROSS-SECTIONAL FRACTURE OF CHILLED LATHE-BED

closeness to actual size with which this is reproduced. A comparison with the $^1/_{64}$-in. graduations on the scale shows the extremely close even grain of this chilled surface.

54 The exact effect of the chill on a casting is well illustrated in Fig. 2. This shows the cross-sectional fracture through the finished ways and a portion of the side wall of the chilled bed for a 30-in. lathe. The reproduction clearly shows that the iron even far away from the chill in the side wall of the casting is dense and of as close grain as other good cast iron; it also shows the much closer grain of

the iron below the finished surface as a result of the chill in closing up the iron and making it harder and more durable. In the 30-in. bed, as plainly shown by this specimen, the close iron produced by the chill extends to a depth of $1\frac{1}{2}$ in. to 2 in. below the finished surface of the ways, thus proving that the chilled iron is not all removed in the planing.

55 *Degree of Hardness.* Scleroscope test on chilled beds finish planed but not scraped gave as a result of eight different tests on four different beds, scleroscope readings of 40 to 42 with an average of 41. Similar tests on chilled beds which had been planed and then scraped gave a practically constant scleroscope reading of 42. Comparative scleroscope tests on a heavy section of unchilled cast iron which would give as nearly as possible conditions parallel to those just quoted gave readings ranging from 18 to 22, with an average of 20.

56 These tests indicate that the chilled ways are made harder than the unchilled. The same is also shown by the planing speed, as we can only plane the chilled beds at a trifle less than half the cutting speed formerly used.

57 In addition to the advantage gained by the hardness, the chilled beds are if anything more uniform than the unchilled. The rough beds as received from the foundry are quite straight, so that the amount of metal removed all along the ways in planing is as nearly uniform as is practicable. Then, too, the scleroscope readings indicated a more nearly constant degree of hardness on the chilled beds than on the unchilled. Fewer castings are unsound than before chills were used.

58 *Action of the Chill.* The chilled surface is produced by a series of cast-iron chill plates each about 6 in. long placed end to end in the mold. The use of separate short plates eliminates much of the warping and twisting which would occur in a long chill plate.

59 If a thick chill plate is used with a low-silicon iron the surface of the casting is chilled so hard that it cannot be machined. The desired result is attained by regulating the thickness of the chill plate to suit the size of the casting for which it is used; then a low-silicon iron of high tensile strength can be successfully poured. The heavier the casting, the thicker the chill plate.

60 The action in the mold is that when the molten iron strikes the cold plate it is chilled and hardened; then the heat in the mass of iron forming the body of the bed casting gradually warms the

chilled surface and the chill plate, thus annealing the casting or "drawing the chill," just as when in tempering a chisel the heat in the shank of the chisel "draws the temper" of the cutting edge to the proper point after the cutting edge has been hardened by quenching in water. This annealing of the chilled surface of the casting produces the desired form of hard, close-grained gray iron.

61 The thickness of the chill plate used is such that the heat in the casting will anneal the surface sufficiently to permit planing, although at a greatly reduced cutting speed, and yet retain the benefits of the chill.

### VALUE OF CHILLED SURFACES

62 The advantages of chilled wearing surfaces for machine tools are:

*a* Much harder surfaces, which experience has proved are vastly more durable than similar unchilled surfaces.

*b* A hard guiding surface with a relatively soft carriage, bringing the bulk of the wear on the carriage and thus maintaining the alignment of the guide.

c *A* denser and much more closely grained surface of the casting, giving better appearance.

*d* An exceptionally smooth finished surface, in which there are no pores where dirt and grit may become imbedded to cause rapid abrasion of the other bearing surface.

63 There are several ways of increasing the durability of working parts, such as by increasing the area of the bearing, by providing more complete lubrication, and by hardening the surfaces. All are successfully used. In general, each method may be used independently of the others. If the areas of the surfaces are as large as special conditions permit, and if the lubrication is thoroughly efficient, there would seem to be no objection to increasing still further the durability by the use of chills.

64 Chilled surfaces are more advantageous on some machines and some parts than on others. In the case of a lathe the carriage will often be used for long periods of time on chuck work or on short jobs between centers which brings all of the wear on a comparatively short length of the bed just in front of the headstock; such uneven wear on the unchilled bed destroys the accuracy of the alignment for long work. Chilling the ways brings the wear principally upon the carriage, and even if the carriage is worn, the alignment at all points along the bed will remain relatively true.

65 Only one manufacturer (see Par. 33) of all who were kind enough to reply on this subject had discontinued the use of chills; the others who are using chills do not report any trouble due to scratching. The objections to the use of chills aside from the one instance quoted, have come from foundrymen who do not and have not used chills.

66 Our own experience, based on the use of chilled ways on beds of all sizes of our lathes for more than two years, is that no internal stresses are created by the chilling; that the surface is not made more susceptible to a peening action; that the surface can be equally as well lubricated as before; that iron of high tensile strength is used: and that the increased hardness and closeness of grain of the chilled surface vastly increases the durability and permanency of alignment. We find no disadvantage except a somewhat increased cost.

### TENSILE STRENGTHS OF VARIOUS IRONS

67 As the letters received from other machine-tool manufacturers do not state tensile strengths, the following statements regarding the general practice of Cincinnati manufacturers are quoted:

68 *A Professor of Mechanical Engineering and Testing:* I believe the general Cincinnati machine-tool practice for good castings runs from 22,000 to 24,000 lb. per sq. in. tensile strength, but owing to the great variety I would not wish to commit myself to any particular figures.

69 *A Cincinnati Chemist:* After tabulating the results of my tests of the tensile strength of cast iron from various sources, I pick some at random to show the average run of machine-tool iron in lb. per sq. in. in this locality: 22,962; 24,090; 24,522; 23,197; 23,260.

70 This will give an idea of what the general run is. Good machineable iron, where the grain does not have to be too close for machine-tool work, should run from 20,000 to 26,000 lb. per sq. in. Low tensile strength is due to too much silicon, sulphur, or phosphorus.

### CONCLUSIONS

71 Chilled surfaces for certain parts are desirable. There might be a limited field for special alloy castings and if any machine-tool manufacturers have experimented with them, the results of their tests would be welcome. There is a wide difference in the chemical analyses of irons used by representative manufacturers—this last circumstance may be due to the different melting conditions in the several foundries; or it may indicate a field where much good could be accomplished by a more complete interchange of information, and by experiments to determine the best mixtures for different purposes.

## DISCUSSION

W. Wallace McKaig called attention to the mixture for heavy lathe of semi-steel, the analysis of which is Si, 1.60, Mn, 0.65 to 0.75, P, 0.40, S, 0.8 to 0.10 (Par. 27), and said that there seemed to be a very wide latitude in the sulphur. Furthermore, an iron with sulphur in the proportion of 0.8 would be white hard all the way through and would be impossible to machine. An iron of the above analysis with sulphur as low as 0.3 would also be white hard. He would appreciate information as to a way of making a 20 per cent semi-steel with sulphur in the proportion of 0.8, in which the castings are soft enough to machine.

A. Lewis Jenkins wrote that most machine tools are required to resist deflections and vibrations, and when sufficiently rigid and massive to satisfy these requirements there is practically no danger of breakage. About the only machine tools designed for strength in which the deflection may be neglected are shears, punches, presses and hammers.

Beds and carriages of lathes; housings, cross-rails, tables and beds of boring mills; columns, and knees of milling machines and shapers; columns, arms and tables of drilling machines, are examples of machine parts requiring stiffness; they never break. The very lowest grades of iron would be entirely satisfactory for these parts from the standpoint of strength, provided the parts were sufficiently heavy to resist the vibrations and deflections caused by the pressure of the tool.

The ultimate tensile or the transverse strength of a test bar is of no greater value in proportioning some parts of machine tools than the specific gravity of the material.

The most important physical property that may be determined by a test is the deflection under a load equal to about $^1/_{10}$ of that required to break the bar, or about 250 lb. at the middle of a 1-in. by 1-in. bar on supports 12 in. apart. It would be better to use a bar that is at least 24 in. between supports in order to get accurate deflection readings.

Hodgkinson's Experimental Researches (1860) gives the ultimate breaking load and the modulus of elasticity at a load of 112 lb. for about 250 bars 1 in. by 1 in. by 54 in. between supports. Plotting

the breaking load against this modulus of elasticity (at 9072 lb. per sq. in. transverse stress) showed no definite relation between the ultimate transverse strength and the elastic properties. It is also evident that no relation exists between the deflections at 112 lb. and at rupture.

In view of these results neither the ultimate strength nor the ultimate deflection of a test bar is of any value in determining the most essential property of the material.

The values of the moduli of elasticity in Hodgkinson's results vary from 11,539,333 to 22,733,400, which means that if a casting were made from each of these materials one might have a deflection twice as great as the other. The variations in the elastic properties of the iron may cause an appreciable effect on the accuracy of some machine tools.

Complicated castings should preferably be made from metal having a small shrinkage, thereby decreasing the initial stress due to cooling, warping, shrinkage cavities and some of the visible defects commonly seen on finished surfaces.

It is not an easy matter for some foundries to get a mixture that will produce desirable machining properties and have a grain sufficiently close to take a high finish. Good results in this respect have been accomplished by the use of charcoal, certain grades of coke, steel scrap and ferro-alloys.

In the selection of castings the machine-tool builder should consider, together with the machining and finishing properties, that property of the iron which resists deflection and upon which depends the stiffness and corresponding accuracy. The ultimate strength is a minor consideration.

Thomas D. West referred to the difficulty experienced by one manufacturer who had replied to the inquiries sent out by the author that he had had trouble from the scratching of chilled ways used on machine tools through some little particle of metal which had settled on them (Par. 33). Mr. West suggested that the chilled surfaces might have been wavy or streaked so as not to present a smooth homogeneous surface; and that the trouble might be due to the failure to use a varnish to coat the chillers with.

All chillers generally require to be coated with wet blacking which must be dried; oils, chalk or other coverings that do not require any heat drying are also used. For the purpose of latheways and chiller

plates or blocks the following recipe is the best the writer knows of and has not to his knowledge been heretofore published. It is what some of our leading carwheel makers use to obtain the perfect surface seen on the threads of their wheels which wear smooth without causing any scratches.

Suitable proportions for this varnish are shellac, 34 lb., lamp black, 4 lb., wood alcohol, 9 gal.; or if a smaller quantity is desired the mixture in the same proportions. The alcohol is put into a barrel with the shellac to cut it up well, so that it will be entirely liquid

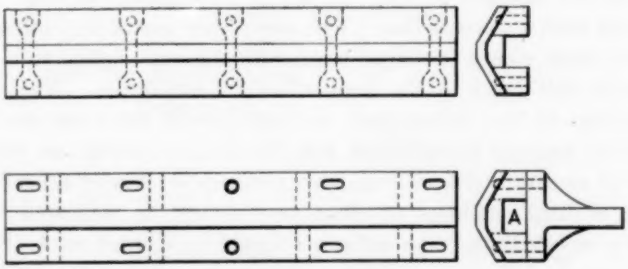



Fig. 3 Chiller for Latheways

before putting in the lamp black. This alcohol costs 50 cents per gal. and is 95 per cent alcohol. Recent experimenting has led to the use of mineral blacking or dust, obtained from foundry companies, in preference to the lamp black, as it gives a smoother skin to the chilled face, does not gum the chillers as the lamp black does, and only about half as much is required.

Should the face of the chillers show signs of any roughness or scale other than that caused by the varnish, some dry sand is rubbed over it with waste, or emery or sand paper may be used. The varnish is applied with a brush, and the chillers are given one coat and then allowed to dry in the open for twenty to thirty minutes. On top of this coat of varnish a thin coating is applied, composed of half kerosene oil and half common black oil, and on top of this again a little mineral blacking is dusted on to prevent the oil and kerosene mixture from forming streaks and to keep it in a smooth body. The dusting is done by shaking it through a hand bag, such as is used by stove molders and general foundrymen for shaking general dry blackings on their molds. This last coat requires only about 15

minutes to dry, after which the mold having the chillers can be poured.

The author mentions the use of a series of cast-iron chiller plates, placed end to end in the mold to eliminate the warping and twisting which is liable to occur in a long chiller plate. Mr. West, however, believed that foundrymen would prefer to use one continuous chiller piece, provided its warping could be prevented, and suggested that where long chillers are desirable, they be made no thicker than necessary to give the required depth of chill, and that means be arranged for bolting a chiller plate to a rigid casting or beam of sufficient stiffness and strength to keep the chiller plate from warping. Fig. 3 indicates such a construction. The upper view shows the chiller plate and the lower view a T-shaped beam with the plate bolted to it. The elongated bolt holes in the beam allow for expansion. The bottom of the lugs of the chiller plate and the face of the beam should be planed to unusual straightness and the chiller casting can be made either of steel or wrought iron, since there is practically no difference in the chilling qualities of these two metals, as indicated by the writer's experiments.[1] An air space should be allowed at *A* (Fig. 3) which may be left open or filled with rammed sand or other material of low heat absorbing quality; or water or air pipes may be inserted in this space to retard a "drawing of the chill" and to eliminate the possibility of the beam becoming heated to such an extent as to cause its warping.

Forrest R. Jones sent the following excerpt from his note book as of value in relation to the durability of flat rubbing surfaces of cast iron. The stationary member is referred to as the "guide," and the moving member as the "block."

Alfred Herbert, Ltd., Coventry, England. Visited September 5, 1902.

Experiments had been made on flat rubbing surfaces of various kinds of cast iron. Both surfaces always of the same iron. The bearing surface of the guide a narrow strip along each side; block held in place by the upper lips of the guide (Fig. 4). Middle of guide cut away for clearance. Oil pocket at each end of guide. Block has narrow strips across its length; ends of cross-strips rest on the bearing strips of the block.

[1] New Processes for Chilling and Hardening Cast Iron, Trans.Am.Soc.M.E., vol. 34, p. 235.

No oil-grooves or indentations (other than the pores of the metal) in the bearing surfaces. Enough oil used to act as a bath.

Each block was run (slipped) back and forth 84 miles on its own particular guide. Hardness was tested by drilling.

Softest metal by drill test showed best wearing qualities. The softer metals were the more porous.

After 84 miles of slipping the softer metals showed practically no wear; the scraper marks still remained on both block and guide. The oil remained clear. Harder blocks showed considerable wear. With them the oil soon became black, almost as black as ink.

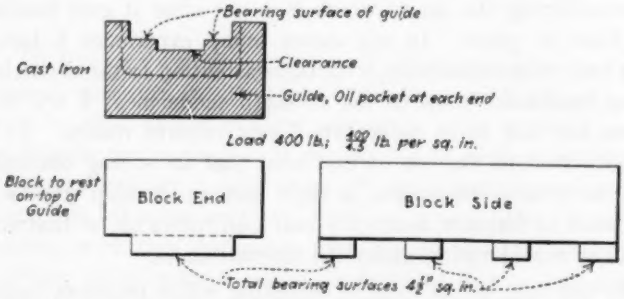



FIG. 4 BLOCK AND GUIDE FOR TESTING RUBBING SURFACES

Total bearing surface of block, $4\frac{1}{2}$ sq. in.; total load, 400 lb.,

$$\text{bearing pressure} = \frac{400}{4.5} = 89 \text{ lb. per sq. in. nearly.}$$

At a date some eight or ten years earlier than that of this memorandum the Aermotor Company of Chicago found that a thoroughly chilled cast-iron bearing would far outlast any other of numerous materials used for the bearings of the horizontal shaft of the vane wheel of their windmills. These bearings were generally neglected as to oiling, and before the chilled cast iron was adopted, gave trouble by rapid wearing out when thus neglected. The cast iron used for the aermotor bearings was chilled hard and white, like a car wheel, a roll for a steel mill or a flour mill, or the mold-board and point of an agricultural plow. The shaft was of ordinary machine steel.

A few years ago, the writer had occasion to test the durability of chilled cast iron as a bearing surface when rubbing against tempered spring steel as the mating bearing surface. The coiled spring gripped

the entire circumference of the round cast-iron member. The cast iron used was chilled hard and white. After the surfaces had slipped over each other more than 40 miles actually measured, and probably half as much more not measured, under all conditions of lubrication and the lack of it, at a bearing pressure more than 300 lb. per sq. in., the chilled iron still showed the marks of the fine-grained emery wheel that had been used to grind its friction surface.

E. H. Mumford (written). Speaking of this question wholly from the foundry standpoint, it occurs to me that too much stress is commonly laid on the strength detail of cast iron for machine tool uses, considering the use to which it is put after it gets into service, other than in gears. In my steam pump experience I have used Keep's test quite extensively, with more attention to the shrinking and chilling tendencies than to the strength tendencies; I try to use a uniform test bar so as to get absolutely relative results. It is undoubtedly true in the use of cast iron that its strong tendencies to shrink in certain dimensions in rigid castings develop stresses which lead almost to fracture before the real load comes on, so that strength is a minor consideration under the circumstances.

I do not refer to the form of shrinking which produces honeycomb spots in the center of joining parts as where pump diaphragms join the side-walls, or spokes join wheel rims and hubs, and which are known as "shrinks" or "draws" in the foundry, for that is a different matter. Castings are weakened more by cooling stresses due to shrinkage than by lack of intrinsic strength as shown by breaking tests, either transverse or tensile.

I had supposed that it was well known that such chilling as Mr. Wood refers to in his paper is entirely different from that in car-wheel or roll or cylinder castings, but the comment made by manufacturer No. 34 seems to show that there is some misconception of what chilling means in the sense in which Mr. Wood has used it in his paper. It should be clearly understood that the chilling referred to bears no relation to the chilling of car wheels or rolls made from low silicon iron.

The quick cooling of high silicon iron on running against a chill prevents the segregation of graphitic carbon in large masses and holds the iron to a closer texture without any such hardening of the surface as interferes with proper machine cutting; whereas, with low silicon iron, the surface, and for considerable depth below it, contains no

graphitic carbon whatever and can be touched only by a grinding wheel or the special chilled tools used in roll turning.

Bradley Stoughton. I think the Society should be congratulated upon this paper, and upon the method which was followed in writing it, whereby the opinion is obtained of several men who have had experience in these matters, and who are willing to give anonymously the benefit of their experience.

I want to call Mr. Wood's attention to the natural alloys of iron, nickel and chromium smelted from the Mayari ores of Cuba. I think Mr. Wood will find that they have been used to some extent in machine tool work.

A. L. De Leeuw (written). I want to call attention to the following statement contained in the paper: "The ideal casting is, of course, one that is so close as not to show any grain when finished, and at the same time be just as hard as it can be, and be worked into shape." I want to warn against the "of course," which seems to take it for granted that we all agree with it. I believe it is a little bit early to agree entirely with this statement. We like to have the appearance of the casting close-grained, but it is doubtful if the close-grained casting is always the most serviceable. Experiences of other people, and some of my own, have shown that the presence of graphite in the casting is of decided benefit; especially if the graphite is near the top after the scale has been removed. This may be the reason why soft blocks wear better than hard ones.

Another thing touched upon in the previous discussion is that it may be advisable to have different hardnesses in the surfaces which wear together; Mr. Wood indicated that it is well to have the lathe bed harder than the carriage. We cannot get away from the wear. The number of cases where this wear is so small as to be negligible are so rare that we may just as well make the general statement that we must provide for the natural wear of moving machine parts. I would suggest that Mr. Wood's statement be broadened out, and that we say that we should make the part which guides, harder than the part which is guided.

The point which was brought out in the discussion in the preceding paper on gears is, that in the softer cast-iron gears there is naturally a wearing at the point of the teeth, and that when this point is worn sufficiently, the wear will then come naturally on that part of the tooth where it should be. I wish to call attention here to

the fact that if the gear is of such softness that the point wears away, then, after the pressure has come to the center, the center will wear away, and then the pressure will come on the point again. I really do not believe that any such kind of action takes place, but rather that there is an undue wear over the entire tooth if the material is too soft. I believe that in order to avoid the excessive wear in gears, the proportions and sizes of the bearing surfaces, and the hardness of the materials, should be as ample as we can make them.

There are certainly a great many pieces where chilled castings can be used to great advantage, and not only for wearing surfaces. This refers especially to T-slots. In most machines, practically all except the very largest, T-slots are finished out, which makes the under side of the lip of the T-slot rather soft. Chilling this T-slot will condense the material, and prevent the lip of the T-slot from swelling unduly when fixtures or pieces of work are clamped down.

D. J. Riggs wrote that the author has referred to chilled iron as that which cannot be planed, filed or scraped; and this is the common conception amongst us. But we all know that some foundry irons will hardly chill at all and that other irons may be found anywhere between these two extremes. It seems that in applying chills to lathe beds, the author has not gone to the extreme limit of hardness. In the case cited where there was trouble from the scratching of chilled ways, they have probably either used an iron that chilled too easily, or used too heavy a chill. After getting the iron too hard they have undoubtedly resorted to the use of loose emery and left some of it imbedded in the iron to make trouble later.

The author did not desire to present a closure.—Editor.

## No. 1417

# A RECORD OF PRESSED FITS

By C. F. MacGill, St. Louis, Mo.

Member of the Society

Articles on forced or pressed fits have been published in various technical journals of the country during the past ten or twelve years, one by Stanley H. Moore[1] appearing in the Transactions of the Society. In but one of the papers noticed is there any reference to the diameter and length of the hub into which the shaft is forced or the material of which the hub is made, three very important elements in a forced fit. The great difference in allowances recommended, and in the pressures shown in the articles referred to, led me to have an exact record kept of each forced fit, from which were later calculated the tension stress, radial pressure and the coefficient of friction. Table 1 forming part of this paper is the result.

2 In my own experience, covering about 20 years, in charge of shops where forced fits were made, I have found that it was not necessary to increase the allowance with the diameter of the shaft, as the increased surface area of the fit added sufficient friction to bring the pressure up to the required tonnage, and that an allowance of from 0.002 in. to 0.004 in. on steel shafts pressed into steel hubs, and an allowance of from 0.003 in. to 0.005 in. on steel shafts pressed into cast-iron hubs of ordinary hardness, gave good results. I have yet to learn of one of these shafts coming loose.

3 There is no doubt in my mind that allowances greater than 0.006 in. on steel shafts pressed into cast-iron hubs, not only do not serve any useful purpose, but tend to set up strains that are injurious to the casting.

4 One large plant with which I am familiar issues an allowance table for pressed fits, the allowance gradually increasing with the shaft diameter. This is not followed in their shaft department, but instead a flat allowance of 0.003 in. is used without regard to the diameter of the shaft.

[1] Fits and Fitting, Trans. Am. Soc. M. E., vol. 24, p. 1158.

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

5 The allowances given by Mr. Moore, in his paper already referred to, are very much too great, while those given by Mr. Riddell[1] in his discussion of the paper are too low for sizes under 12 in.

6 According to Mr. Moore's formula $A = 2D+0.5$ and shown in his table, the allowance for an 8-in. shaft would be $8 \times 2+0.5=$ 0.0165 in., and on a 12-in. shaft $12 \times 2+0.5=0.0245$ in., and as, from Table 1 of actual fits, an allowance of 0.002 in. on 8-in. shafts pressed into steel hubs showed pressures of from 75 to 110 tons and an allowance of 0.004 in. on a 12-in. shaft pressed into a cast-iron hub showed 130 tons, it does not seem that greater allowances are either necessary or advisable.

7 The records in Table 1 cover 206 fits of diameters varying in size from 3.5 in. to 20 in., sufficient in range and number to demonstrate the correctness and value of the practice followed. All of these measurements and gage readings were taken by the same inspector, and as he was a thoroughly reliable man, I am satisfied that they are correct. The records are all of fits made on electric generators and motors.

8 The table shows the diameter of the shaft, the diameter of the bore, the length of the seat, the diameter of the hub, the material of which the hub is made, the allowance (the difference between the diameter of the shaft and the diameter of the bore of the hub into which the shaft is pressed), the pressure in tons required to force the shaft in, the maximum tension stress in the bore in pounds per square inch, the radial pressure on the surface of the shaft in pounds per square inch, and the coefficient of friction. These figures are given as a record of actual experience on sizes between the diameters shown. They are not claimed to represent correct practice beyond these dimensions.

9 In Table 1 the last three columns were figured from formulae developed by Prof. A. Morley and published in Engineering, August 11, 1911. In these formulae

$R_1$ = outer radius of hub
$R_2$ = inner radius of hub = radius of shaft after fitting
$A = R_1 - R_2$ = thickness of hub wall
$D = 2R_2$ = diameter of shaft
$P_2$ = radial compressive stress in lb. per sq. in. at radius $R_2$
$f_t$ = hoop tension per sq. in. at radius $R_2$

[1]Trans. Am. Soc. M. E., vol. 24, p. 1173.

$J$ = excess in original diameter of shaft over that of bore of hub = allowance

$$\left.\begin{array}{l} E = \text{Young's modulus for shaft material, assumed 30,000,000} \\ E_1 = \text{Young's modulus for hub material, assumed 15,000,000} \end{array}\right\} \text{thus } E = 2E_1$$

$$\left.\begin{array}{l} \dfrac{1}{m} = \text{Poisson's ratio for shaft material} \\ \dfrac{1}{m_1} = \text{Poisson's ratio for hub material} \end{array}\right\} \text{assumed } m = m_1 = 4$$

Total tension stress at bore of hub

$$f_t = \frac{\dfrac{J}{D} \times E}{\left\{\left(\dfrac{m-1}{m} + \dfrac{1E}{m_1 E_1}\right)\dfrac{R^2_1 - R^2_2}{R^2_1 + R^2_2} + \dfrac{E}{E_1}\right\}}$$

Normal pressure on surface of shaft

$$P_2 = \frac{\dfrac{J}{D} - \dfrac{f_t}{E_1}}{\dfrac{m-1}{m\,E} + \dfrac{1}{m_1\,E_1}}$$

Coefficient of friction

$$\mu = \frac{P}{P_2}$$

where

$P_2$ = total normal pressure in tons

$P$ = pressure in tons required to force shaft into hub

EXAMPLE RECORD NO. 84

Cast-iron hub on steel shaft $D = 6.0025$ in., $J = 0.0025$ in., length of hub $l = 7$ in., thickness of hub $t = 3\,5/8$ in. Assuming $m = m_1 = 4$ and $E = 2E_1 = 30,000,000$, then

TABLE 1 DATA AND CALCULATIONS ON PRESSED FITS

| Number of Record | Diameter of Shaft, In. | Diameter of Bore, In. | Length of Seat, In. | Diameter of Hub, In. | Material in Hub | Allowance, In. | Pressure, Tons | Maximum Tension Stress of Bore, Lb. per Sq. In. | Radial Pressure on Surface of Shaft, Lb. per Sq. In. | Coefficient of Friction |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 3.504 | 3.500 | 6 | 5 | Steel | 0.004 | 30 | 25520 | 8680 | 0.105 |
| 2 | 3.504 | 3.500 | 6 | 5 | Steel | 0.004 | 45 | 25520 | 8680 | 0.157 |
| 3 | 3.504 | 3.500 | 6 | 5 | Steel | 0.004 | 45 | 25520 | 8680 | 0.157 |
| 4 | 3.503 | 3.500 | 6 | 5 | Steel | 0.003 | 50 | 19185 | 6525 | 0.233 |
| 5 | 3.504 | 3.500 | 6 | 4½ | Steel | 0.004 | 30 | 27360 | 6840 | 0.133 |
| 6 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 7 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 8 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 40 | 21125 | 8875 | 0.120 |
| 9 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 10 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 40 | 21125 | 8875 | 0.120 |
| 11 | 4.003 | 4.000 | 8 | 5½ | Steel | 0.003 | 30 | 17175 | 5325 | 0.112 |
| 12 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 13 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 14 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 45 | 21125 | 8875 | 0.134 |
| 15 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 16 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 33 | 21125 | 8875 | 0.0985 |
| 17 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 35 | 21125 | 8875 | 0.104 |
| 18 | 4.004 | 4.000 | 8 | 5½ | Steel | 0.004 | 40 | 22900 | 7100 | 0.112 |
| 19 | 4.004 | 4.000 | 6 | 7¼ | Steel | 0.004 | 50 | 19545 | 10456 | 0.127 |
| 20 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 45 | 21125 | 8875 | 0.103 |
| 21 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 50 | 21125 | 8875 | 0.149 |
| 22 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 50 | 21125 | 8875 | 0.149 |
| 23 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 45 | 21125 | 8875 | 0.103 |
| 24 | 4.004 | 4.000 | 6 | 6¼ | Steel | 0.004 | 45 | 21125 | 8875 | 0.103 |
| 25 | 4.054 | 4.051 | 6 | 6¼ | Steel | 0.003 | 40 | 15915 | 6286 | 0.153 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 26 | 4.054 | 4.051 | 6 | 6¼ | Steel | 0.003 | 40 | 15915 | 6286 | 0.153 |
| 27 | 4.505 | 4.500 | 8 | 6½ | Steel | 0.005 | 60 | 24665 | 8665 | 0.122 |
| 28 | 4.505 | 4.500 | 8 | 6½ | Steel | 0.005 | 50 | 24665 | 8665 | 0.102 |
| 29 | 4.5035 | 4.500 | 8 | 6½ | Steel | 0.0035 | 45 | 17285 | 6050 | 0.131 |
| 30 | 4.505 | 4.503 | 6½ | 5¾ | Cast Iron | 0.002 | 20 | 5750 | 1455 | 0.3 |
| 31 | 4.503 | 4.500 | 6 | 10 | Cast Iron | 0.003 | 30 | 7070 | 4690 | 0.151 |
| 32 | 4.503 | 4.500 | 6 | 10 | Cast Iron | 0.003 | 30 | 7070 | 4690 | 0.151 |
| 33 | 4.503 | 4.500 | 8 | 6½ | Steel | 0.003 | 40 | 14815 | 5185 | 0.136 |
| 34 | 4.503 | 4.500 | 8 | 6½ | Steel | 0.003 | 40 | 14815 | 5185 | 0.136 |
| 35 | 4.880 | 4.875 | 7 | 7½ | Steel | 0.005 | 48 | 21885 | 8865 | 0.101 |
| 36 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 45 | 22365 | 8385 | 0.117 |
| 37 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 45 | 22365 | 8385 | 0.117 |
| 38 | 4.880 | 4.875 | 7 | 7¼ | Steel | 0.005 | 60 | 22365 | 8385 | 0.143 |
| 39 | 4.880 | 4.875 | 7 | 7¼ | Steel | 0.005 | 45 | 22365 | 8385 | 0.107 |
| 40 | 4.880 | 4.874 | 6 | 7¼ | Steel | 0.006 | 67 | 26835 | 10064 | 0.145 |
| 41 | 4.880 | 4.875 | 7 | 8 | Steel | 0.005 | 40 | 21060 | 9690 | 0.077 |
| 42 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 50 | 22365 | 8385 | 0.130 |
| 43 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 50 | 22365 | 8385 | 0.130 |
| 44 | 4.880 | 4.876 | 7 | 7¼ | Steel | 0.004 | 40 | 17890 | 6710 | 0.111 |
| 45 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 50 | 22365 | 8385 | 0.130 |
| 46 | 4.880 | 4.875 | 7 | 8 | Steel | 0.005 | 55 | 21060 | 9690 | 0.106 |
| 47 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 45 | 22365 | 8385 | 0.117 |
| 48 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 45 | 22365 | 8385 | 0.117 |
| 49 | 4.880 | 4.875 | 6 | 7½ | Steel | 0.005 | 45 | 21885 | 8865 | 0.113 |
| 50 | 4.880 | 4.875 | 6 | 7½ | Steel | 0.005 | 45 | 21885 | 8865 | 0.113 |
| 51 | 4.880 | 4.875 | 6 | 7½ | Steel | 0.005 | 45 | 21885 | 8865 | 0.113 |
| 52 | 4.880 | 4.875 | 6 | 7½ | Steel | 0.005 | 45 | 21885 | 8865 | 0.113 |
| 53 | 4.880 | 4.875 | 7 | 8 | Steel | 0.005 | 55 | 21060 | 9690 | 0.106 |
| 54 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 40 | 22365 | 8385 | 0.104 |
| 55 | 4.880 | 4.875 | 6 | 7¼ | Steel | 0.005 | 50 | 22365 | 8385 | 0.130 |

TABLE 1—*Continued*

| Number of Record | Diameter of Shaft, In. | Diameter of Bore, In. | Length of Seat, In. | Diameter of Hub, In. | Material in Hub | Allowance, In. | Pressure, Tons | Maximum Tension Stress of Bore, Lb. per Sq. In | Radial Pressure on Surface of Shaft, Lb. per Sq. In. | Coefficient of Friction |
|---|---|---|---|---|---|---|---|---|---|---|
| 56 | 4.880 | 4.875 | 7 | 8 | Steel | 0.005 | 50 | 21060 | 9690 | 0.096 |
| 57 | 5.003 | 5.000 | 7¾ | 10 | Cast Iron | 0.003 | 25 | 6545 | 3930 | 0.150 |
| 58 | 5.003 | 5.000 | 8 | 10 | Cast Iron | 0.003 | 20 | 6545 | 3930 | 0.081 |
| 59 | 5.002 | 5.000 | 5½ | 7¾ | Steel | 0.002 | 50 | 8500 | 3500 | 0.330 |
| 60 | 5.440 | 5.437 | 9 | 13 3/16 | Cast Iron | 0.003 | 25 | 5730 | 4055 | 0.0806 |
| 61 | 5.440 | 5.437 | 9 | 13 3/16 | Cast Iron | 0.003 | 25 | 5730 | 4055 | 0.0806 |
| 62 | 5.502 | 5.500 | 8 | 13 | Cast Iron | 0.002 | 22 | 3800 | 2650 | 0.120 |
| 63 | 5.502 | 5.500 | 8½ | 13 | Cast Iron | 0.002 | 25 | 3800 | 2650 | 0.128 |
| 64 | 5.630 | 5.625 | 7 | 13 | Cast Iron | 0.005 | 30 | 9340 | 6390 | 0.076 |
| 65 | 5.630 | 5.625 | 7½ | 8¼ | Steel | 0.005 | 40 | 19550 | 7115 | 0.085 |
| 66 | 5.629 | 5.625 | 8¼ | 8¼ | Steel | 0.004 | 65 | 15640 | 5695 | 0.207 |
| 67 | 5.630 | 5.625 | 7½ | 8½ | Steel | 0.005 | 65 | 19185 | 7480 | 0.131 |
| 68 | 5.630 | 5.625 | 7 | 8 | Steel | 0.005 | 45 | 19900 | 6765 | 0.1075 |
| 69 | 5.630 | 5.625 | 7 | 8¼ | Steel | 0.005 | 65 | 19550 | 7115 | 0.148 |
| 70 | 5.630 | 5.625 | 6½ | 8¼ | Steel | 0.005 | 65 | 19550 | 7115 | 0.159 |
| 71 | 5.630 | 5.625 | 7 | 8¼ | Steel | 0.005 | 65 | 19535 | 7115 | 0.147 |
| 72 | 5.630 | 5.625 | 7 | 8½ | Steel | 0.005 | 55 | 19185 | 7480 | 0.119 |
| 73 | 5.630 | 5.625 | 7 | 9¼ | Steel | 0.005 | 48 | 18260 | 8405 | 0.092 |
| 74 | 5.630 | 5.625 | 7½ | 8½ | Steel | 0.005 | 50 | 19185 | 9480 | 0.101 |
| 75 | 5.630 | 5.625 | 7 | 8¼ | Steel | 0.005 | 55 | 19535 | 7115 | 0.125 |
| 76 | 5.630 | 5.625 | 8 | 8 | Steel | 0.005 | 55 | 20000 | 665 | 0.1185 |
| 77 | 5.7525 | 5.750 | 8 | 12¾ | Cast Iron | 0.0025 | 18 | 4610 | 3060 | 0.081 |
| 78 | 5.752 | 5.750 | 8 | 12¾ | Cast Iron | 0.002 | 20 | 3690 | 2375 | 0.117 |
| 79 | 6.005 | 6.000 | 8 | 9 | Steel | 0.005 | 70 | 18055 | 6945 | 0.134 |
| 80 | 6.002 | 6.000 | 8 | 9 | Steel | 0.002 | 55 | 7220 | 2780 | 0.262 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 81 | 6.003 | 6.000 | 7¾ | 13¾ | Cast Iron | 0.003 | 30 | 5265 | 3575 | 0.115 |
| 82 | 6.005 | 6.000 | 8 | 9 | Cast Iron | 0.005 | 30 | 10485 | 3225 | 0.123 |
| 83 | 6.003 | 6.000 | 8 | 14½ | Cast Iron | 0.003 | 40 | 5200 | 3680 | 0.144 |
| 84 | 6.0025 | 6.000 | 7 | 13¼ | Cast Iron | 0.0025 | 40 | 4425 | 2920 | 0.207 |
| 85 | 6.0025 | 6.000 | 8 | 14½ | Cast Iron | 0.0025 | 38 | 4425 | 2920 | 0.173 |
| 86 | 6.130 | 6.125 | 7 | 8¼ | Steel | 0.005 | 65 | 19065 | 5475 | 0.176 |
| 87 | 6.253 | 6.250 | 9 | 11½ | Steel | 0.003 | 60 | 19065 | 5475 | 0.134 |
| 88 | 6.253 | 6.250 | 9 | 11½ | Steel | 0.003 | 65 | 9320 | 5080 | 0.145 |
| 89 | 6.253 | 6.250 | 9 | 11½ | Steel | 0.003 | 65 | 9320 | 5080 | 0.145 |
| 90 | 6.2735 | 6.273 | 9 | 11½ | Steel | 0.0005 | 15 | 1550 | 8401 | 0.20 |
| 91 | 6.5025 | 6.500 | 8 | 21¾ | Cast Iron | 0.0025 | 40 | 3790 | 3170 | 0.155 |
| 92 | 6.503 | 6.500 | 8 | 14¾ | Cast Iron | 0.003 | 40 | 4871 | 3280 | 0.149 |
| 93 | 6.502 | 6.500 | 8 | 14¾ | Cast Iron | 0.002 | 38 | 3245 | 2190 | 0.213 |
| 94 | 6.503 | 6.500 | 6 | 12 | Cast Iron | 0.003 | 40 | 5160 | 2835 | 0.23 |
| 95 | 6.503 | 6.500 | 8 | 13½ | Cast Iron | 0.003 | 35 | 4980 | 3110 | 0.134 |
| 96 | 6.502 | 6.500 | 8 | 13 | Cast Iron | 0.002 | 25 | 3355 | 2115 | 0.152 |
| 97 | 6.503 | 6.500 | 8 | 11¼ | Steel | 0.003 | 90 | 9220 | 4610 | 0.24 |
| 98 | 6.5025 | 6.500 | 8 | 14¾ | Cast Iron | 0.0025 | 25 | 4060 | 2735 | 0.112 |
| 99 | 6.503 | 6.500 | 8 | 14¾ | Cast Iron | 0.003 | 35 | 4800 | 3395 | 0.126 |
| 100 | 6.502 | 6.500 | 12 | 10 | Cast Iron | 0.002 | 50 | 3680 | 1495 | 0.275 |
| 101 | 6.610 | 6.605 | 11 | 14 | Steel | 0.003 | 60 | 8000 | 5620 | 0.094 |
| 102 | 6.939 | 6.937 | 7¾ | 13 | Cast Iron | 0.002 | 15 | 3210 | 1785 | 0.1 |
| 103 | 6.939 | 6.937 | 8 | 11 | Cast Iron | 0.002 | 17 | 3410 | 1465 | 0.133 |
| 104 | 7.002 | 7.000 | 10 | 12 | Steel | 0.002 | 75 | 5730 | 2820 | 0.242 |
| 105 | 7.002 | 7.000 | 10 | 12 | Steel | 0.002 | 80 | 5730 | 2820 | 0.268 |
| 106 | 7.003 | 7.000 | 10 | 12½ | Steel | 0.003 | 90 | 8325 | 4515 | 0.181 |
| 107 | 7.002 | 7.000 | 9 | 17 | Cast Iron | 0.002 | 50 | 2965 | 2110 | 0.24 |
| 108 | 7.003 | 7.000 | 10 | 13 | Steel | 0.003 | 85 | 8285 | 4615 | 0.168 |
| 109 | 7.003 | 7.000 | 10 | 13 | Steel | 0.003 | 80 | 8285 | 4615 | 0.158 |
| 110 | 7.003 | 7.000 | 9 | 16¾ | Cast Iron | 0.003 | 40 | 4465 | 3135 | 0.129 |

TABLE 1—*Continued*

| Number of Record | Diameter of Shaft, In. | Diameter of Bore, In. | Length of Seat, In. | Diameter of Hub, In. | Material in Hub | Allowance, In. | Pressure, Tons | Maximum Tension Stress of Bore, Lb. per Sq. In. | Radial Pressure on Surface of Shaft, Lb. per Sq. In. | Coefficient of Friction |
|---|---|---|---|---|---|---|---|---|---|---|
| 111 | 7.002 | 7.000 | 10 | 12 | Steel | 0.002 | 95 | 5730 | 2820 | 0.307 |
| 112 | 7.003 | 7.000 | 8 | 12¾ | Cast Iron | 0.003 | 35 | 4815 | 2600 | 0.153 |
| 113 | 7.003 | 7.000 | 10 | 13 | Steel | 0.003 | 50 | 8285 | 4615 | 0.0985 |
| 114 | 7.0035 | 7.000 | 8 | 14¾ | Cast Iron | 0.0035 | 45 | 5375 | 3400 | 0.12 |
| 115 | 7.003 | 7.000 | 9 | 17 | Cast Iron | 0.003 | 50 | 4450 | 3160 | 0.18 |
| 116 | 7.003 | 7.000 | 8 | 12 | Steel | 0.003 | 100 | 8605 | 4235 | 0.269 |
| 117 | 7.034 | 7.031 | 8 | 10½ | Steel | 0.003 | 75 | 9260 | 3520 | 0.262 |
| 118 | 7.034 | 7.031 | 8 | 10½ | Steel | 0.003 | 35 | 9260 | 3520 | 0.113 |
| 119 | 7.1895 | 7.187 | 7 | 13 | Cast Iron | 0.0025 | 25 | 3915 | 2085 | 0.152 |
| 120 | 7.2525 | 7.2495 | 9 | 16¾ | Cast Iron | 0.003 | 35 | 4350 | 2475 | 0.115 |
| 121 | 7.437 | 7.435 | 7½ | 11 | Cast Iron | 0.002 | 25 | 3275 | 1220 | 0.234 |
| 122 | 7.4395 | 7.437 | 8 | 13 | Cast Iron | 0.0025 | 25 | 3810 | 1970 | 0.136 |
| 123 | 7.502 | 7.499 | 8 | 12¾ | Cast Iron | 0.003 | 55 | 4605 | 2230 | 0.238 |
| 124 | 7.5025 | 7.500 | 8 | 13 | Cast Iron | 0.0025 | 35 | 3810 | 1905 | 0.194 |
| 125 | 7.5027 | 7.500 | 8 | 14½ | Cast Iron | 0.0027 | 20 | 3970 | 2290 | 0.0925 |
| 126 | 7.5029 | 7.501 | 8 | 25½ | Cast Iron | 0.0019 | 15 | 2460 | 2145 | 0.074 |
| 127 | 7.534 | 7.531 | 10 | 12 | Steel | 0.003 | 50 | 8320 | 3620 | 0.117 |
| 128 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 60 | 5015 | 2485 | 0.202 |
| 129 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 60 | 5015 | 2485 | 0.202 |
| 130 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 60 | 5015 | 2485 | 0.202 |
| 131 | 8.002 | 8.000 | 16 | 14 | Steel | 0.002 | 110 | 4975 | 2525 | 0.217 |
| 132 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 80 | 5015 | 2485 | 0.27 |
| 133 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 85 | 5015 | 2485 | 0.286 |
| 134 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 75 | 5015 | 2485 | 0.253 |
| 135 | 8.002 | 8.000 | 9½ | 13¾ | Steel | 0.002 | 80 | 5015 | 2485 | 0.27 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 136 | 8.002 | 8.000 | 9 | 13½ | Steel | 0.002 | 80 | 5065 | 2435 | 0.318 |
| 137 | 8.003 | 8.000 | 9 | 17 | Cast Iron | 0.003 | 50 | 4025 | 2565 | 0.173 |
| 138 | 8.4395 | 8.4375 | 7 | 11 | Cast Iron | 0.002 | 25 | 3060 | 795 | 0.34 |
| 139 | 8.4395 | 8.437 | 8 | 11 | Cast Iron | 0.0025 | 18 | 3820 | 833 | 0.205 |
| 140 | 8.440 | 8.4375 | 8 | 11 | Cast Iron | 0.0025 | 15 | 3820 | 833 | 0.17 |
| 141 | 8.503 | 8.500 | 8 | 14¾ | Cast Iron | 0.003 | 35 | 4035 | 2015 | 0.162 |
| 142 | 8.503 | 8.500 | 8 | 14½ | Cast Iron | 0.003 | 45 | 4055 | 1980 | 0.212 |
| 143 | 8.503 | 8.500 | 9 | 15 | Cast Iron | 0.003 | 40 | 3815 | 2370 | 0.141 |
| 144 | 8.503 | 8.500 | 8 | 14¾ | Cast Iron | 0.003 | 45 | 4435 | 2020 | 0.208 |
| 145 | 8.5025 | 8.500 | 6½ | 15 | Cast Iron | 0.0025 | 30 | 3000 | 2255 | 0.154 |
| 146 | 8.505 | 8.5015 | 8 | 15 | Cast Iron | 0.0045 | 45 | 6005 | 3085 | 0.136 |
| 147 | 8.507 | 8.504 | 9 | 11 | Cast Iron | 0.003 | 40 | 4565 | 1145 | 0.292 |
| 148 | 8.564 | 8.562 | 14 | 13½ | Cast Iron | 0.002 | 30 | 2765 | 848 | 0.1875 |
| 149 | 8.939 | 8.937 | 8 | 13 | Cast Iron | 0.002 | 18 | 2860 | 762 | 0.211 |
| 150 | 8.939 | 8.937 | 8 | 13 | Cast Iron | 0.002 | 20 | 2860 | 762 | 0.234 |
| 151 | 8.939 | 8.937 | 8 | 13 | Cast Iron | 0.002 | 22 | 2860 | 762 | 0.257 |
| 152 | 9.002 | 8.999 | 12 | 20½ | Cast Iron | 0.003 | 65 | 3515 | 2375 | 0.161 |
| 153 | 9.002 | 8.999 | 12 | 20½ | Cast Iron | 0.003 | 65 | 3515 | 2375 | 0.161 |
| 154 | 9.0025 | 8.999 | 12 | 20½ | Cast Iron | 0.0035 | 75 | 4100 | 2375 | 0.159 |
| 155 | 9.0025 | 8.999 | 12 | 20½ | Cast Iron | 0.0035 | 75 | 4100 | 2375 | 0.159 |
| 156 | 9.0015 | 8.9995 | 9 | 16¾ | Cast Iron | 0.002 | 22 | 2480 | 1370 | 0.126 |
| 157 | 9.002 | 9.000 | 10 | 14 | Steel | 0.002 | 80 | 4710 | 1955 | 0.29 |
| 158 | 9.002 | 9.000 | 10 | 14 | Steel | 0.002 | 75 | 4710 | 1955 | 0.272 |
| 159 | 9.003 | 9.000 | 12 | 20¾ | Cast Iron | 0.003 | 70 | 3500 | 2395 | 0.171 |
| 160 | 9.003 | 9.000 | 12 | 20¾ | Cast Iron | 0.003 | 70 | 3500 | 2395 | 0.171 |
| 161 | 9.003 | 9.000 | 12 | 21 | Cast Iron | 0.003 | 70 | 3495 | 2410 | 0.171 |
| 162 | 9.003 | 9.000 | 12 | 20¾ | Cast Iron | 0.003 | 70 | 3500 | 2395 | 0.171 |
| 163 | 9.002 | 9.000 | 13 | 20½ | Cast Iron | 0.002 | 75 | 2345 | 1585 | 0.257 |
| 164 | 9.003 | 9.000 | 12 | 20¾ | Cast Iron | 0.003 | 90 | 3500 | 2395 | 0.22 |
| 165 | 9.003 | 9.000 | 12 | 14¾ | Cast Iron | 0.003 | 85 | 3890 | 1775 | 0.298 |

TABLE 1—*Continued*

| Number of Record | Diameter of Shaft, In. | Diameter of Bore, In. | Length of Seat, In. | Diameter of Hub, In. | Material in Hub | Allowance, In. | Pressure, Tons | Maximum Tension Stress of Bore, Lb. per Sq. In. | Radial Pressure on Surface of Shaft, Lb. per Sq. In. | Coefficient of Friction |
|---|---|---|---|---|---|---|---|---|---|---|
| 166 | 9.003 | 9.000 | 10 | 18⅜ | Steel | 0.003 | 100 | 6200 | 3795 | 0.187 |
| 167 | 9.003 | 9.000 | 10 | 18⅜ | Steel | 0.003 | 80 | 6200 | 3795 | 0.149 |
| 168 | 9.003 | 9.000 | 9 | 17 | Cast Iron | 0.003 | 45 | 3700 | 2080 | 0.17 |
| 169 | 9.003 | 9.000 | 9 | 17 | Cast Iron | 0.003 | 50 | 3700 | 2080 | 0.189 |
| 170 | 9.939 | 9.937 | 7½ | 13 | Cast Iron | 0.002 | 25 | 2745 | 720 | 0.295 |
| 171 | 9.9998 | 9.996 | 9 | 16¾ | Cast Iron | 0.0038 | 60 | 4400 | 2080 | 0.204 |
| 172 | 10.003 | 10.000 | 9 | 16¾ | Cast Iron | 0.003 | 40 | 3495 | 1640 | 0.172 |
| 173 | 10.004 | 10.000 | 12 | 21 | Cast Iron | 0.004 | 90 | 4205 | 2715 | 0.176 |
| 174 | 10.0035 | 10.000 | 14 | 21 | Cast Iron | 0.0035 | 80 | 3765 | 2375 | 0.153 |
| 175 | 10.002 | 10.000 | 12 | 18 | Steel | 0.002 | 55 | 3925 | 2475 | 0.128 |
| 176 | 10.502 | 10.500 | 8 | 13 | Cast Iron | 0.002 | 16 | 2505 | 550 | 0.22 |
| 177 | 11.002 | 11.000 | 12⅝ | 18 | Steel | 0.002 | 70 | 3750 | 1705 | 0.189 |
| 178 | 11.002 | 11.000 | 12⅝ | 18 | Steel | 0.002 | 80 | 2750 | 1705 | 0.216 |
| 179 | 11.005 | 10.9997 | 14 | 20¾ | Cast Iron | 0.0053 | 38 | 5350 | 3005 | 0.0525 |
| 180 | 11.002 | 11.000 | 12¾ | 18 | Steel | 0.002 | 85 | 3750 | 1705 | 0.24 |
| 181 | 11.002 | 11.000 | 13½ | 18 | Steel | 0.002 | 90 | 3750 | 1705 | 0.227 |
| 182 | 11.002 | 11.000 | 13 | 18 | Steel | 0.002 | 90 | 3750 | 1705 | 0.235 |
| 183 | 11.002 | 11.000 | 13 | 18 | Steel | 0.002 | 70 | 2750 | 1705 | 0.183 |
| 184 | 11.002 | 11.000 | 12¾ | 18 | Steel | 0.002 | 90 | 3750 | 1705 | 0.24 |
| 185 | 11.002 | 11.000 | 13 | 18 | Steel | 0.002 | 100 | 3750 | 1705 | 0.261 |
| 186 | 11.0025 | 11.000 | 10 | 17½ | Cast Iron | 0.0025 | 50 | 2680 | 1160 | 0.25 |
| 187 | 11.0035 | 11.000 | 10 | 17½ | Cast Iron | 0.0035 | 60 | 2755 | 1625 | 0.214 |
| 188 | 11.064 | 11.062 | 12 | 18¾ | Cast Iron | 0.002 | 48 | 2100 | 975 | 0.236 |
| 189 | 12.004 | 12.000 | 12 | 20¾ | Cast Iron | 0.004 | 130 | 3815 | 1895 | 0.302 |
| 190 | 12.003 | 12.000 | 13 | 20½ | Cast Iron | 0.003 | 70 | 2875 | 1405 | 0.204 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 191 | 12.004 | 12.000 | 13 | 21 | Cast Iron | 0.004 | 78 | 3795 | 1930 | 0.165 |
| 192 | 13.0045 | 13.002 | 13 | 21½ | Cast Iron | 0.0025 | 100 | 2235 | 1035 | 0.362 |
| 193 | 13.003 | 13.000 | 14 | 22 | Cast Iron | 0.003 | 85 | 2660 | 1265 | 0.235 |
| 194 | 13.0045 | 13.000 | 12 | 22 | Cast Iron | 0.0045 | 90 | 4000 | 1905 | 0.193 |
| 195 | 13.0045 | 13.000 | 12 | 22 | Cast Iron | 0.0045 | 100 | 4000 | 1905 | 0.213 |
| 196 | 13.003 | 13.000 | 12 | 25½ | Cast Iron | 0.003 | 70 | 2530 | 1470 | 0.195 |
| 197 | 13.004 | 13.000 | 13 | 26 | Cast Iron | 0.004 | 100 | 3355 | 2015 | 0.187 |
| 198 | 13.0035 | 13.000 | 13 | 22 | Cast Iron | 0.0035 | 110 | 3105 | 1490 | 0.252 |
| 199 | 13.003 | 13.000 | 12¾ | 21 | Steel | 0.003 | 125 | 4790 | 2130 | 0.225 |
| 200 | 14.033 | 14.031 | 14 | 21 | Steel | 0.003 | 150 | 4785 | 1845 | 0.264 |
| 201 | 14.033 | 14.031 | 14 | 21 | Steel | 0.002 | 100 | 3075 | 1185 | 0.274 |
| 202 | 14.5035 | 14.500 | 13 | 25½ | Cast Iron | 0.0035 | 100 | 2720 | 1430 | 0.235 |
| 203 | 16.004 | 16.000 | 14 | 25½ | Cast Iron | 0.004 | 120 | 2865 | 1415 | 0.241 |
| 204 | 20.002 | 20.000 | 20 | 28⅞ | Steel | 0.002 | 160 | 2220 | 780 | 0.327 |
| 205 | 20.0025 | 19.999 | 20 | 29 | Steel | 0.0026 | 140 | 2880 | 1020 | 0.218 |
| 206 | 20.0025 | 19.999 | 20 | 29 | Steel | 0.0026 | 140 | 2880 | 1020 | 0.218 |

$$\text{Max. tension stress } f_t = \frac{\frac{0.0025}{6} \times 30{,}000{,}000}{\left(\frac{3}{4}+\frac{1}{4}\times 2\right)\frac{44-9}{44+9}+2} = 4425 \text{ lb. per sq.in.}$$

$$\text{Radial pressure } p_2 = \frac{\frac{0.0025}{6} - \frac{4425}{15{,}000{,}000}}{\frac{3}{4\times 30{,}000{,}000}+\frac{1}{4\times 15{,}000{,}000}} = 2920 \text{ lb.per sq.in.}$$

$$\text{Coefficient of friction} = \frac{40}{\frac{\pi\times 6\times 7\times 2920}{2000}} = 0.207$$

## DISCUSSION

John E. Sweet. During the last third of a century at the works of the Straight Line Engine Company, Syracuse, N. Y., we have had experience on a kind of forced fits different from that of Mr. MacGill, and with results that will, I fancy, surprise him and others.

Our crankshafts were made by using two flywheels with very large bosses for the throws of the crank, and, in forcing the crankpins in these bosses out of the center, and in forcing in two pieces to form the shafts. These shafts and cranks were of Coffin treated steel, forced into the cast-iron bosses, and these forced fits were without keys. The size of the crankpins varied from 3⅞ to 5 15/16 in., and the shafts varied from 3⅜ to 4¾ in. The allowance for the fits was after we learned the trade, about 0.0015 in. to the inch, and another thing we learned was that while the boring and turning was very carefully done there would be a little wearing away of the ridges at the parts that first entered, so that to get a rigid bearing at the shoulder, the fit of the shaft was given 0.001-in. taper.

To give some idea of the permanence of these connections, we turned out thousands of these fits, and, in cases where 4¾-in. shaft fits without keys were coupled to generator shafts, where the shafts in the armatures were 6 in. or more, and with 1½-in. keys, the fits

in the wheels never slipped. Where the holes are reamed and the shafts are ground, two-thirds as much allowance is ample, and, where the shaft and boss are both steel, it is likely that one-half as much allowance will do, and that is dependent on the length of the fit whether turned and bored or reamed and ground fits.

Ought not a shaft 8 in. in diameter take a pressure four times as great to make a good force fit as one 4 in. in diameter? Will the same allowance give this? Mr. MacGill's table only shows twice as much. There is four times as much surface in the 8-in. size as in the 4-in., even though only the same length of fit.

Sanford A. Moss. Mr. MacGill's mathematical methods are undoubtedly correct. However, his values of force for pressing and the corresponding values of coefficients of friction seem very high. There is great variation in the individual cases of the table, but this always occurs. I have many records similar to those of Mr. MacGill's and they vary as much. These are for steel hubs on steel shafts and I have given their general average in a paper on this subject.[1] This led to a coefficient of friction of about 0.038, which is to be compared with Mr. MacGill's average for steel hubs of about 0.12, a difference of 300 per cent. To take actual cases, the average of a great many tests of pressure to force car wheels on axles with about 5-in. bore and 6-in. length of hub, with an allowance of one mil per inch, is about 20 tons, and with an allowance of two mils per inch is about 40 tons. Mr. MacGill's table includes a great many cases of steel hubs from about 4.5 in. to 5.5 in. with an allowance of one mil per inch and his average pressure is about 50 tons. In the paper previously alluded to, an average test figure was given for force to press a rather thick steel hub, of 1560 lb. per inch of bore and per inch of hub length, for an allowance of one mil per inch. This gives for the above wheels with one mil per inch, about 23 tons. I know of some cases where values about 40 tons have been actually obtained, and they have been ascribed to inaccuracies of machine work of hub and shaft.

The test on which the figure 1569 was based and those giving the value of 20 tons force were with ground shafts and carefully reamed holes. A shaft or bore with ordinary machinery may be out of round or out of straightness or have other irregularities amounting to several thousandths.

[1] Increase of Bore of High-Speed Wheels by Centrifugal Stresses, Sanford A. Moss, Trans. Am. Soc. M. E., vol. 34, p. 909.

Possibly Mr. MacGill's coefficients and forces are for ordinary machinery, and the lower values are for cases, with greater accuracy. The lower figures given are for lubrication with white lead or something similar. It is to be presumed that Mr. MacGill used a lubricant.

The formulae and methods used in the paper already quoted are identical with those of Mr. MacGill. In the writer's paper the case taken was for a solid or hollow shaft with a steel hub. Mr. MacGill takes a solid shaft with hub of any material, and values for a steel hub equivalent to the above are obtained by using steel coefficient. Mr. MacGill used 0.25 for Poisson's ratio. The value is probably nearer 0.30, but the effect is very small.

Mr. MacGill rightly concludes that large force fit allowances are not necessary in usual cases. However, in cases where they are necessary, as in high-speed wheels, they cannot be objected to because of excessive hub stresses unless the formulae show large stresses. For proportionately thick hub, the stresses with one mil per inch of bore allowance, are the same regardless of diameter.

It is to be noticed that the maximum effective stress in the hub is not simply the tangential stress, but is shown by the maximum shear theory to be the sum of the tangential and radial compressive stress. This is $\frac{EJ}{D}$ when shaft and hub are of the same material. The sum of Mr. MacGill's stresses will be found to be $\frac{EJ}{D}$ for steel. It may also be noted that the ratio of the radial and tangential stresses when hub and shaft are alike is

$$\frac{R_1^2-R_2^2}{R_1^2+R_2^2}$$

Harry M. Lane wrote that his company's records comprise data upon about 10,000 pressed fits, which go back about 20 years, and he had found it gratifying to note that they embrace all the data possible, there being 24 facts recorded with regard to each pressed fit, including diameter and length of hub into which the shaft is forced, and material of which the hub is made, area of fitted surface and the pressure at the first, second, third and fourth quarters of the length of the fit, which show the effect as recorded by the pressure of varying lengths

of fit in each particular case without requiring comparison of different tests.

Before making these records they sometimes had trouble with loose cranks and crankpins, but since beginning the records these troubles practically ceased.

In common with others, they used to attempt additional security in the fastening of crankpins by riveting the pin over at the back of the crank, by screwing it up at the back with a thin, fine-threaded nut or by driving a taper key through slots in the crankpin and its boss.

Of course, if the pin does not fit the hole, none of these makeshifts are effective, and if the crankpin is a fit, they are unnecessary. For last twenty years it has been the company's practice to make a dependable fit of the crankpin and let it go at that.

John Riddell spoke of the necessity experienced in the practice of the General Electric Company for varying the allowance for fits in order to meet the conditions existing. In the case of turbine engines it might be necessary in some cases to make a much greater allowance, due to the high speed at which the turbine was to operate, with consequent expansion of the wheel, than in another case where the turbine was to run at slower speed. As a specific instance, on a shaft 23 in. in diameter, an allowance of 0.020 in. was made for the fit. On this particular piece of apparatus it was calculated that when it revolved at 20 per cent above normal speed, the bore would become 0.017 in. larger, or in other words, there would be 0.003 in. left for fit at this speed.

This phase of the subject was treated by Mr. Sanford A. Moss[1] of the company's works at Lynn, Mass., who has gone into the matter carefully and treated it mathematically.

A. B. Carhart (written). It may be proper to suggest that in making calculations for forced fits, the surface of the contact area is a very important factor. The formulæ assume contact of the entire area of the shaft and of the bore. We all know how seldom this is actually realized in practice, because of the irregularities of any machine operations. In railroad shops, in the pressing on of car wheels it is surprising to see the approximation to accurate size and taper that will sometimes pass for satisfactory work and be forced into a fit. I cannot believe that in some cases any considerable part of the

[1] Increase of Bore of High-Speed Wheels by Centrifugal Stresses, Trans. Am. Soc. M. E., vol. 34, p. 895.

total area is actually in contact; yet some of such forced fits prove themselves to be reliable, durable and dependable in service.

More exact information should be required concerning some forced fits as for example in the pressing of car wheels upon axles for use in transportation service. I believe that formerly there was ground for complaint concerning wheels that would either work loose or would split in the hubs, and the manufacturers were called upon to replace wheels claimed to be defective or to pay the expense resulting from failure. Consequently it became of great importance to them to show that the workmanship in any case was satisfactory and that the fit was proper. It was necessary to know that there was proper area of contact, that sufficient power was applied to insure a permanent job, and especially that the pressure was not so excessive as to cause strain within the hub. It is as serious and as troublesome to have car wheels crack from over pressure in making pressed fits, as it is to have them run loose; either fault is a serious one.

Consequently it is now common practice to have a record made in the shop of each fit at the time it is pressed on. Special recording gages are used, which automatically trace a diagram of the pressure applied at each stage of the fit, indicating at the same time the forward movement of the ram of the press that forces the axle into the hub of the wheel. The record begins when the first contact is made, where the pressure begins to be appreciable, and as the axle moves forward, in pressing on the wheel, or in any shaft fitting, the actual gage pressure is recorded at each inch of movement, until the wheel comes to the shoulder or to the final position, and the maximum thrust applied there is also recorded. Such a chart record can be filed away, marked with the number of the job and of the wheel, and the shop has proof, in the future, of a proper and accurate fit, or of any sudden rise in the pressure at any point owing to a jam due to irregularity of the taper or to inaccuracy of the diameters concerned in making the fit.

The Author. The original object of my paper was to prove that "it is not necessary and may be dangerous to increase the allowance on pressed fits, with the diameter of the shaft." As stated, the tension stress, radial pressure and coefficient of friction were calculated some time after the records of the fits were made. The calculations were made primarily to check the correctness of my claim that flat allowances of from 0.002 in. to 0.004 in. on steel shafts in steel hubs, and 0.003 in. to 0.005 in. on steel shafts in cast-iron hubs, without regard

to the diameter, gave good results, and that allowances greatly in excess of these may prove dangerous; and secondly, to arrive at a proper basis for establishing standard force fit allowances for ordinary machine shop practice, where high centrifugal forces and extremes in temperature need not be taken into consideration. In my judgment the results of these calculations uphold my claim.

I am pleased that the formulae used meet the approval of Mr. Moss. For the sake of simplicity Poisson's ratio was taken at 0.25 for both cast iron and steel, since the use of different values has no appreciable influence on the result. The value of $E$ for cast iron was kept constant at 15,000,000 partly for the same reason, and partly because where greater allowances are used and $E$ diminishes, these allowances will bring the cast iron into the region of dangerous stresses, especially if hard key driving should be added to excessive allowances.

As to the coefficient of friction it is somewhat difficult to draw reliable conclusions from the figures obtained by the formula used.

Mr. Lane is the author of the paper referred to in the following quotation: "In but one of the papers noticed is there any reference to the diameter and length of the hub into which the shaft is forced or the material of which the hub is made."

In answer to Professor Sweet's question:, "Ought not a shaft 8 in. in diameter take pressure four times as great to make a good force fit as one 4 in. in diameter? Will the same allowance give this?" I think not, and my records show that it does not. If this were the case an allowance proportionally less would be logical.

In all cases the surfaces of the fits covered by my table were lubricated with white lead and linseed oil before being pressed together. Too much care cannot be taken in making the surfaces of the fit true and smooth and in the keeping of an exact record of each fit made, possibly with a diagram showing the pressure at different stages as suggested by Mr. Carhart and Mr. Lane.

## No. 1418

# A NEW PROCESS OF CLEANING PRODUCER GAS

By H. F. Smith, Lexington, Ohio

Member of the Society

In 1902 the writer instituted a series of investigations to determine the nature of the mechanical impurities present in producer gas from bituminous coal with a view to devising more effective methods for their removal. These investigations have since been continued and have resulted in the development of a commercial apparatus involving some new and interesting principles.

2 The tar and other mechanical impurities present in raw bituminous producer gas are in an extreme state of subdivision. The number of particles present is so great and the quantity of gas to be handled in commercial plants so large that the problem presents more than ordinary difficulties. The effectiveness of the ordinary types of mechanical gas washers and purifiers leaves much to be desired. The primary object has accordingly been to produce equipment that will be capable of yielding gas of a higher degree of cleanness than obtainable by ordinary methods. The apparatus in its present stage of development can be readily understood from Fig. 1.

3 The raw producer gas on leaving the producer is first cooled to a point where the tar vapors are condensed by being passed through a primary cooler or condenser. From this the gas is carried into an ordinary rotary gas pump *B* which delivers the gas under pressure into the main *C*; it is then delivered through a porous diaphragm *E* and discharged from there into the main *F*. A sump or separator *G* is provided in which the tar accumulates.

4 The structure of the diaphragm *E* is a matter of considerable importance for the successful carrying out of this process and the materials used seem to have an important bearing on the operation of the equipment. The diaphragm must be sufficiently porous to permit the gas and tar to pass freely, otherwise it will soon become blocked with deposits from the gas and fail to operate. Many ma-

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

terials may be used for this purpose, but at present spun glass is preferred. The glass fibers are not only entirely unaltered by chemical action but seem to possess the necessary physical properties for the successful carrying on of this process. The spun glass in the form of ordinary glass wool (which should be carefully distinguished from slag wool, as the latter is not practicable for this purpose) is built into the form of a uniform diaphragm and is retained between two

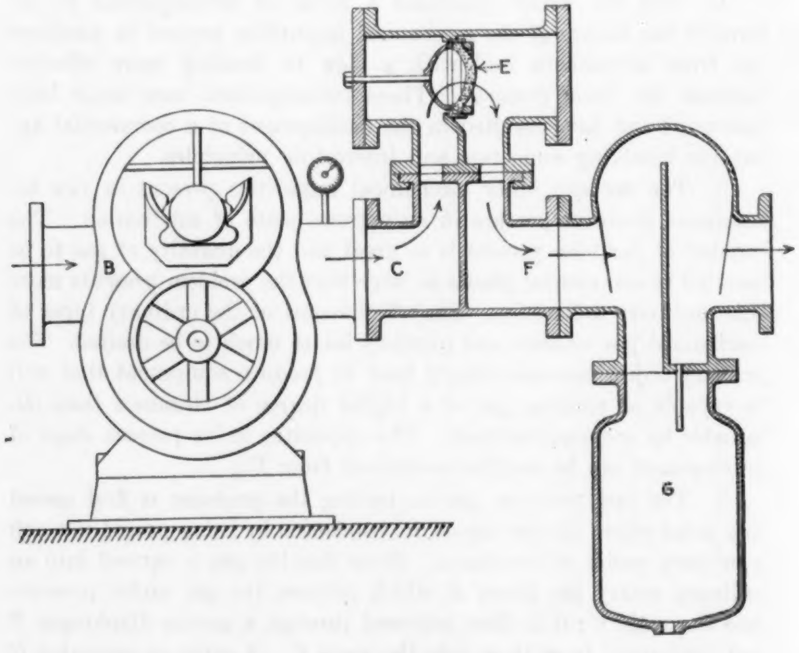



Fig. 1 Static Scrubber Layout

metal screens. The density of the diaphragm can be regulated by the quantity of glass used and by the degree of compression maintained between the metal screens. Ordinarily, this diaphragm is made up to a thickness of approximately one-quarter of an inch. The diameter of the diaphragm must be adjusted in accordance with the quantity of gas to be treated. Ordinarily, about 400 cu. ft. per hr. can be handled for each square inch of diaphragm area. No tar is retained in the diaphragm, both tar and gas being discharged together.

5 In passing the diaphragm an important change in the physical state of the tar occurs. On the entering side the tar exists in a large number of minute particles, ordinarily known as tar fog. In passing the diaphragm these particles are caused to coalesce so that on the discharge side the tar particles are of relatively large dimensions, so large in fact that they can no longer be carried forward in the gas current and immediately separate out by gravity. All that is necessary for the complete separation of the tar from the gas is to provide a sump, or drip, into which the precipitated tar can drain.

6 It appears to be possible to secure almost any desired degree of gas cleanness simply by regulating the pressure maintained across the diaphragm. In ordinary commercial operation, it is found that a difference in pressure of from $2\frac{1}{2}$ to 4 lb. will give a degree of gas cleanness that is ample for any commercial requirement. Thirty cubic feet of gas cleaned in this way can be passed through a white filter paper without producing any discoloration.

7 The distinction between this process and the process of purification by filtration can be best shown by outlining the conditions essential for each process:

*a* In filtration the best separation is secured when the rate at which the materials to be separated pass the filtering medium is slow. One of the substances to be separated remains in the filter.

*b* In the process in question good results can be secured only when the velocity of the gas passing through the diaphragm is very high. Nothing whatever remains in the diaphragm.

8 At low velocities the gas will pass through the porous diaphragm used in this apparatus without any apparent alteration, and the degree of effectiveness of cleaning is directly related to the velocity of flow. For example, the degree of cleanness produced with the velocity of flow resulting from 5-lb. pressure is very much greater than the degree of cleanness produced by the velocity of flow resulting from 1-lb. pressure, and when the velocities are as low as those produced by a pressure of a few ounces only, there is no perceptible change in the tar content of the gas after passing through the diaphragm.

9 No water is used in connection with this process except that required to cool the gas. As a consequence there is no production of

tar emulsion and the water flows from the condenser perfectly clear. The tar separated by this process is practically water free, and can accordingly be used for any purpose to which coal tar is adaptable. One sample of tar drawn directly from the receiver showed on distillation a water content of less than 1 per cent as compared with from 20 to 60 per cent which is ordinarily present in gas producer tar from mechanical washers. The calorific value of producer tar from Hocking coal is approximately 15,800 B.t.u. per lb., about 140,000 B.t.u. per gal.

10 For the maintenance of continuous operation the tar must be sufficiently fluid to pass through the porous diaphragm without creating undue resistance, and therefore it is necessary to maintain the temperature of the gas entering the diaphragm at a point that will reduce the viscosity of the tar to as low a point as is consistent with complete condensation of the tar vapors.

11 It is also apparent that this apparatus would not be well suited to use on gas containing large quantities of lamp black or for the purification of gas from coals yielding very heavy viscous tars. For high volatile coals, however, such as are found in Ohio, Indiana and Illinois, and for lignite, it has been found in practice to be thoroughly practical and effective. It is possible that further developments may extend the applicability of this method to conditions which are not now considered practical.

12 The exact method by which this tar extractor operates has not been conclusively demonstrated. Two theories have been advanced which may possibly cover the ground: The first and most obvious is that the tar particles are precipitated by being brought into direct collision with the threads or filaments of the porous diaphragm.

13 That this does not constitute a complete explanation of the process is indicated by the fact that the material of which the porous diaphragm is constructed has a marked bearing on the effectiveness of the process and would indicate some action other than simple mechanical collision. For example, if the porous diaphragm is made up of steel wool instead of glass wool (the physical structure of the diaphragm being as nearly as possible the same in each case) the process does not operate with anything like the effectiveness secured with glass diaphragms. It would seem that the possibility for collision would be the same in both cases.

14 A phenomenon, first observed by the writer in 1902 during some experimental investigations, gives further credence to the theory

that there is some action other than pure mechanical collision. If the gas is caused to pass through a small tube with perfectly smooth walls, as for example a tube of glass, no particular precipitation of tar occurs as long as the velocities of travel are slow. However, as the velocities increase to a point where there is considerable friction between the gas and the surface of the containing tube a heavy precipitation of tar occurs on the surface of the glass. This fact leads to the conclusion that friction is in some way concerned in this process, since the probability of mechanical collision is rather remote. Since friction between rapidly moving gases and enclosing tubes is known to be productive of electrical phenomena, it was assumed that this might possibly have some bearing on the action of this process. In fact this interpretation was the one first placed upon the phenomenon observed in 1902 and an effort was made to work out a tar extractor along this line.

15 An experimental apparatus was constructed at that time in which heavily charged electrodes were employed to precipitate the tar particles and it was found that fairly effective results could be secured. Experiments along this line continued for a number of years, but the difficulties in the way of producing commercially practical apparatus caused its final abandonment. The rate at which the tar particles could be moved through the gas under the influence of moderate potential gradients was very slow. It was accordingly necessary to use exceedingly high potentials in order to secure effective results. With the spacing of electrodes of approximately 1½ in. a potential difference of 25,000 to 35,000 volts was required for effective precipitation. On account of the difficulty of maintaining proper insulation under these potentials and on account of the great danger of serious injury to an unskilled operator in manipulating apparatus of this kind, this method was not considered practical. It was noted, however, that by decreasing the distance between the electrodes a very marked decrease in potential was observed. Accordingly another experiment was devised which will perhaps throw still further light on the method of operation of the process under consideration.

16 A series of electrodes was prepared with exceedingly small intervening spaces, and placed in connection with a source of direct electro motive force, the potential difference between the plates being much below that required to produce any ionizing discharge. It was found that at these small distances distinct cleaning effects could be

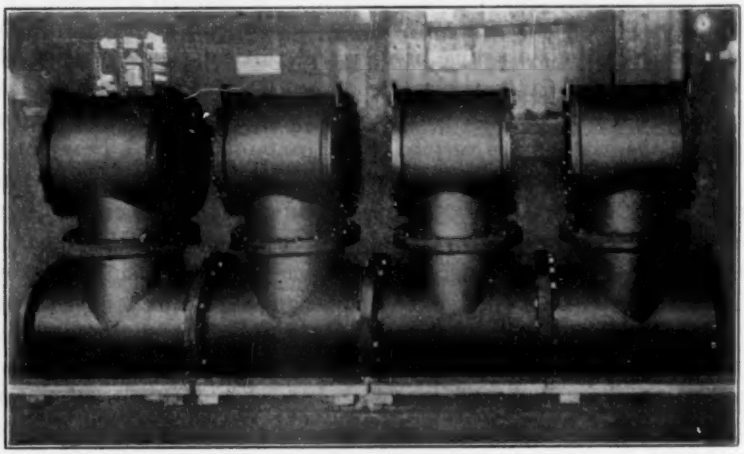


FIG. 2 SMITH TYPE F TAR EXTRACTOR. CAPACITY 250,000 CU. FT. PER HOUR

FIG. 3 VIEW SHOWING VARIOUS PARTS OF COMMERCIAL EXTRACTOR

obtained without ionization. Maintenance of the electric charge from external sources was troublesome owing to electrolytic short circuits occurring between the electrodes through deposits of tar and moisture from the gas. The fact, however, that there is a distinct attraction exerted by electrified plates at comparatively low potentials (which is sufficient to cause a precipitation of tar particles from gases) leads to the conclusion that if the distance between the electrodes could be reduced sufficiently the potential differences required for effective electrical action would be very small.

17 It would seem possible, therefore, that in addition to the effects of mechanical collision there might be a distinct electrical attraction exerted by the glass fibers constituting the porous diaphragm which are located at microscopic distances from each other and which are undoubtedly subjected to some electrification from friction with the gas currents. If the possibility of such electrical action is considered, the increased effectiveness of glass as compared with steel for the construction of the porous diaphragm is satisfactorily explained. That friction of this nature is capable of producing electrical disturbances of considerable magnitude is well established.

18 It might be interesting in this connection to refer to recent experimental determinations by Professor Dolezalek of Charlottenberg, Germany, who showed that the friction of liquid benzol against the sides of containing pipes may set up potential differences of as much as 3000 volts. As a very small fraction of this potential would be sufficient to account for the observed effects, it would seem only reasonable to presume that in addition to the effects of mechanical collision which undoubtedly exist, there is some electrical action which is of material aid in causing the coalescence of the particles of tar fog.

19 Whatever may be the correct explanation of the phenomenon, its effectiveness and practical importance are beyond question. The first commercial equipment of this kind has now been in continuous operation for approximately 18 months. This outfit is handling gas for a producer gas power plant of approximately 1000-h.p. capacity. The second commercial equipment is handling approximately 900-h.p. producer gas, and has been in daily service for approximately ten months. The largest single installation is an equipment for cleaning 200,000 cu. ft. of gas per hour. An idea of the dimensions and form of this equipment can be obtained from Figs. 2 and 3. This installa-

tion is operated in connection with a single producer unit which is of interest because it is one of the largest single unit plants ever installed in this country. This producer has an effective grate surface of 250 sq. ft. and is rated to gasify 3000 lb. of Illinois bituminous coal per hour.

## DISCUSSION

F. R. Hutton said that the author had brought out an important fact, that the particles on the discharge side are very much larger and the product is very much less an emulsion than on the upstream side of the porous diaphragm.

The suggestion of electrical action might also be supplemented by the reference in another paragraph to the fact that at certain velocities there is considerable friction between the gas and the surfaces over which it passes, which causes a heavy precipitation of tar on the surface of the glass. If he had added cohesion of the particles he thought the idea might have been expressed more fully.

Another striking feature of the author's presentation is the significance of spun glass as compared with any other medium. This means that it is not a filtration, as the particles are not left on the filtering or separating diaphragm. It provides for a cleansing of the diaphragm.

Wm. T. Magruder said he did not believe the author had considered colloidal action as a third reason for the action of the spun glass. He referred to the wonderful transformations that take place in organic chemistry by this action.

R. H. Fernald. In regard to the power required, in reply to a question by Mr. H. J. K. Freyn, I have no authentic figures, but the operators at the plants where this particular type of extractor is installed claim that the power required is much less than in the old type centrifugal tar extractor, one operator putting the amount at about 25 or 30 per cent of that required for the old type.

Regarding the claims of cleanliness advanced by Mr. Smith, I cannot make any specific comment other than to say that for some reason the operators find it desirable to change this spun glass at intervals of perhaps three to six weeks. This change is, however, a very simple one and can be made at any time in a few moments and the expense involved is a mere trifle. The frequency of these changes

of the spun glass seems to vary with the operator and the interval between changes speedily increases as the operator becomes more familiar with this type of extractor. In one or two of the installations which I recently visited, the operators changed the spun glass at intervals of one or two weeks when the plant was first started. The time was soon extended to three weeks and later to six weeks. Whether the interval will ever be extended to a year is a point that I can not at present answer. In one or two of the plants using this type of extractor, the tar is found to be sufficiently free from water and of such quality as to be of commercial value. In one installation the engineer is credited with $1.20 per bbl. for the tar which he uses in other processes in the plant. It seems to be a tar of real commercial value.

Inquiry of the operators at the plants indicated that the material is comparatively inexpensive, and that comparatively little of the material has to be used, approximately one pound. It was a comparatively inexpensive item, but I do not know where the material is secured.

Some idea of the proportions of this tar extractor may be had if I state that the extractor shown by the author has an overall height of about 5 ft. from the base of the gas main to the top of the upper tee, that is, they are about 30-in. tees. As this plant was operated when I visited it, three of the tees were in series and the fourth was cut out. The tee that was turned at right angles to the main was entirely out of service, and it was then a simple matter to take off the cap and get at the grid containing the spun glass inside. The process of changing the material and cleansing the tar extractor is seen to be very simple.

It may be of additional interest to know that in one installation recently visited which has been operated for several years, considerable difficulty has been experienced from tar in the engines while using the centrifugal tar extractor, according to the statement of the operator. This same operator states that since changing to the new type of extractor outlined in the paper, some two months ago, they have had no trouble from tar in the engine, and they have had a great deal of satisfaction from this extractor.

This question of tar and other impurities from producer gas in the cylinder of the engine is a very important one. I recently visited a producer-gas installation of 4000 h.p. The same company has another plant which I understand is of the same size, operating on

natural gas. The statement of the company is to the effect that the monthly cylinder oil bill is very much more with the producer-gas engine than with the natural gas engine. This brings to our attention some important factors concerning the impurities in producer gas and the necessity of careful cleansing.

The Author did not desire to present a closure—EDITOR.

No. 1419

# PRESENT STATUS OF THE LARGE GAS ENGINE IN EUROPE

By Prof. P. Langer, Aachen, Germany

Member of the Society

The tendency to utilize in gas engines the enormous quantities of waste gases from blast furnaces had its inception in Germany about 20 years ago. The attempt was then made to create units of larger power by increasing in size the parts of smaller motors and by using a larger number of cylinders. The two-cycle system also seemed to offer a suitable method of operation for large engines, on account of its more efficient utilization of the mechanism as compared with the single-acting four-cycle system. Full success was not attained, however, until about 11 years ago when the large gas engine was brought to a high state of perfection by the Maschinenfabrik Nuernberg in the form of the double-acting four-cycle type, with two cylinders arranged in tandem. It was recognized that the principles on which to base the design of large gas engines should involve (*a*) the greatest possible accessibility of all parts exposed to the gases of combustion, and (*b*) relieving the cylinder wall of the weight of the piston.

2 These principles have not only been proven to be correct but their observance has been found to be a necessary condition for the success of large gas engines.

3 The gas engine puts much higher demands upon the attendant than any other prime mover. It creates its own potential energy by conversion of the chemically latent heat energy of the gas. Almost any defect of the machine, inherent or acquired, which interferes with the conversion of the chemical energy, acts destructively upon the machine, not to mention its influence upon the power output. The simple consideration that gas is being burnt without rendering its equivalent in power, leads to the conclusion that the balance is being transmitted to the cooling water under pressures and temperatures beyond what are permissible, and that exhaust gases unallowably hot

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

are flowing around the exhaust valves. In the case of the steam engine a leaky piston does no further damage than to increase the steam consumption. The governor automatically adjusts for a longer cut-off and the engine pulls through as long as the boiler furnishes sufficient steam. With the gas engine, however, a leaky piston causes ignition at the wrong time which means a release of heat energy at a time and place when this energy can do no useful work. A continuation of the operation under such conditions is not permissible. Any attempt to force the operation would cause heavy damage to the machine.

4 It is the duty of the designer to face these facts and to design the machine in such a manner that it is possible to give it the careful attention needed to prevent such disturbances. Such attention cannot be given when access to such vital parts as pistons, stuffing box packings and valves can be had only after a long and difficult job of disassembling. The author knows of an engine which, after six years of continuous day and night operation, showed only 0.015 in. wear in the diameter of the cylinder. This success, although immediately traceable to the careful attention this machine received, is nevertheless indirectly due to the designer who made it possible to give such attention through the excellence of his design.

5 An instance of defective design in this respect is the case of a stuffing box in which it is possible to replace the packing rings only by unscrewing the joint between the piston rod and crosshead. The packing rings must be split and the whole packing must be easily removable in order that a new packing may be put in during a brief interruption of the operation. The result of neglecting this seemingly minor point of design is a continuation of operation with leaky packings and danger of warping the piston rods.

6 A further basic condition for uninterrupted operation is pure materials, pure gas and pure cooling water.

7 Operation with impure materials puts demands upon the operator which it is impossible to meet, except by considerable reduction of the output on account of the long interruptions required for cleaning. It is evident that the economy of the plant suffers materially through reduction of the output, as the proportion of the unproductive capital, the idle machinery, increases.

8 The attempt to attain stratification of the mixture inside of the cylinder has led to very complicated valve gears. It was hoped that a gas valve, which remained open during only a part of the suction stroke would direct the gas in such a manner that a combustible mixture would be present at the point of ignition even under the lightest loads, while the balance of the combustion space would be filled with inert air. The result, however, did not justify this hope. Instead of stratification there was only a bad mixture resulting in irregular and uneven operation. Today the hope of attaining stratification can be considered as being finally disposed of and designers have returned to the simple throttling valve gear.

9 Throttling of gas and air simultaneously, or in other words, regulation of the quantity of mixture only, is to be preferred to the throttling of gas only, as the former method makes possible a more certain control of the power developed.

10 Quantity regulation will give satisfaction only if the action of the governor upon the throttle valve has been given careful consideration. Present and older designs that are faulty in this respect are frequently seen. It is wrong to let the governor act in such a way that the valve opening is proportional to the effective travel of the governor sleeve, as the following consideration will show:

11 The result of throttling is to reduce the quantity of mixture drawn into the cylinder during each stroke, on account of the reduced openings for gas and air. The volume of the charge remains the same, as the cylinder is always completely filled. The density of the charge will become less and correspondingly its weight and the amount of energy supplied.

12 These relations are shown in the curves of Fig. 1. Starting with a certain velocity $C$ of the mixture in the valve, which is determined by the opening of the valve, the curve shows the velocity as a function of the position of the governor. This velocity can be produced only by a certain drop in pressure as the mixture passes to the cylinder, so that the law according to which the absolute pressure in the cylinder changes is also definitely determined.

13 The density of the charge is proportional to the absolute pressure and the absolute weight of the charge of constant volume is proportional to the density, and so is the amount of heat energy supplied. The shape of this curve shows that the lower part of the

travel of the governor sleeve is almost without influence and that the total regulation is limited to the upper travel.

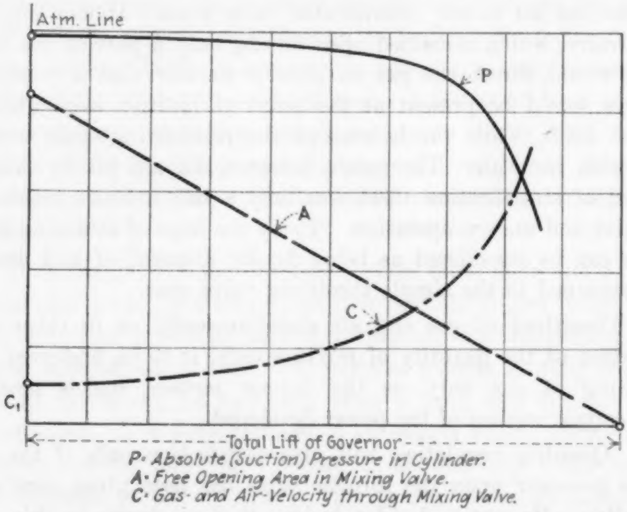



**Fig. 1** Curve showing Relations of Opening of Mixing Valve, under Control of Governor, to Suction in Cylinder and Velocity through Valve

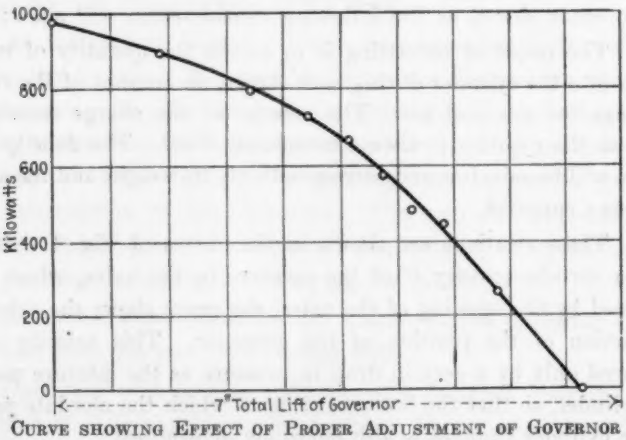



**Fig. 2** Curve showing Effect of Proper Adjustment of Governor Mechanism to Obtain Uniform Utilization of Total Lift of Valve

14 The result of this wrong regulation is marked irregularity of the indicator diagrams, as the smallest motion of the governor causes considerable changes in the quantity of mixture supplied. Combustion

is irregular, there is tendency to backfire, or in one word, the machine "does not govern."

15 The remedy is found in changing the connection between the governor sleeve and the throttling mechanism in a manner such that in the low positions of the governor the throttling action is more intensive than in the upper positions. This action can be accomplished in a simple manner by off-setting the connecting link between the governor and the throttling valve, similar to the arrangement found in Corliss valve gears. This allows a uniform utilization of the effective lift of the governor, and consequently stable and quiet regulation, without any complication whatever. Fig. 2 shows a diagram, characterizing

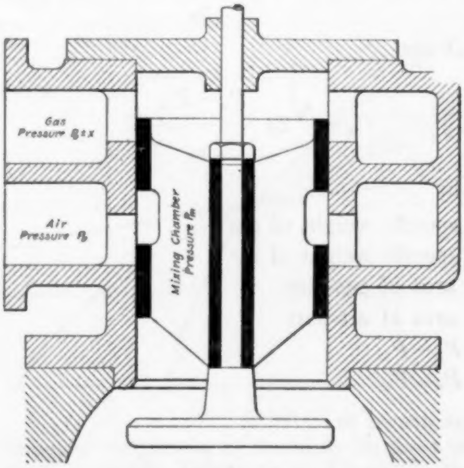



FIG. 3 TYPICAL DESIGN OF MIXING VALVE FOR LARGE GAS ENGINES

the regulation as taken from an engine having a properly designed throttling gear.

16 Even the best considered scheme of regulation will not avail, if the proportion of mixture, that is the ratio of gas to air, is not properly controlled. In engines that have separate pumps for air and gas, as usual in two-cycle machines, where therefore, air and gas are furnished in measured quantities, it is a comparatively easy matter to get the proper proportion, at least at full load. Difficulties are found, however, in four-cycle machines, where gas and air are drawn in in parallel by the working piston. In this case the proportion of mixture is not readily controlled, because the quantities drawn in depend not only on the free opening of the admission valve,

but on the product of this opening area and the velocity of flow through it. Incidental changes in pressure, due to static or dynamic causes, may influence this velocity strongly.

17 If we denote by $P_g$ and $P_a$ the pressure in the gas and air pipes respectively, before they reach the mixing chamber (Fig. 3), and by $P_m$ the pressure prevailing at the same time in the mixing chamber, then $P_g—P_m$ is the difference in pressure creating the velocity of gas and $P_a—P_m$ the difference in pressure creating the velocity of air. Consequently velocity of gas

$$C_g = \sqrt{2g\frac{P_g - P_m}{\gamma_g}}$$

and velocity of air

$$C_a = \sqrt{2g\frac{P_a - P_m}{\gamma_a}}$$

If we denote by

$\gamma_g$ = specific weight of gas
$\gamma_a$ = specific weight of air
$A_g$ = area of gas-port
$A_a$ = area of air-port
$H_g = P_g - P_m$
$H_a = P_a - P_m$

Volume of gas drawn in

$$V_g = A_g \sqrt{2g\frac{H_g}{\gamma_g}}$$

Volume of air drawn in

$$V_a = A_a \sqrt{2g\frac{H_a}{\gamma_a}}$$

Therefore the ratio of mixture

$$M = \frac{V_g}{V_a} = \frac{A_g}{A_a}\sqrt{\frac{\gamma_a}{\gamma_g} \times \frac{H_g}{H_a}}$$

18 It may be noted that the difference in specific gravity must be considered when deciding upon the proportions of the ports for correct mixture in engines that use gases lighter than air. A disregard of

this fact leads to too rich mixtures and the engine becomes choked with gas. The many failures of engines working with coke oven gas are partly explained by wrongly proportioned gas ports. At least, it was found that proper operation resulted in many cases from materially reduced gas ports.

19 If, for the sake of simplicity, we assume in the case of blast furnace gas, the ratio of the port areas as well as the ratio of specific gravities, to equal unity; and if we assume the equation $H_g = H_a \pm x$, where $x$ denotes the difference in gas pressure, as compared with the

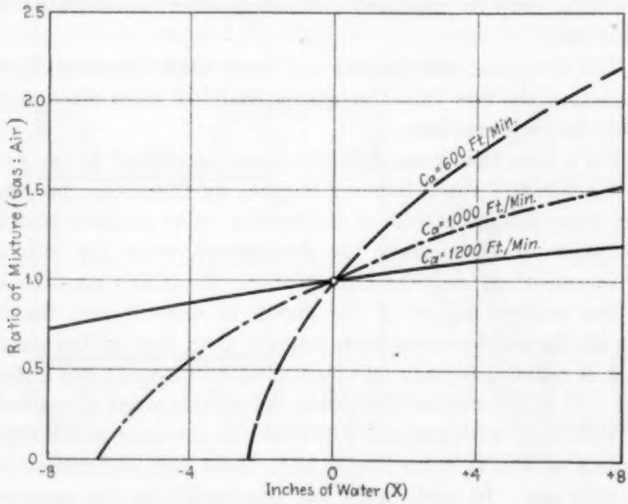



FIG. 4 CURVE SHOWING VARIATIONS IN RATIO OF MIXTURE OF GAS AND AIR WITH DIFFERING VELOCITIES OF AIR DRAWN IN

air pressure, then we have the very simple relation, for the ratio of mixture

$$M = \sqrt{1 \pm \frac{x}{H_a}}$$

This equation shows what means have to be employed in order to keep the variations of the mixture within permissible limits. It shows the necessity of keeping $H_a$ high, which means that we must work with high velocities in the mixing valve, because the larger the value of $H_a$, the smaller the value of $\frac{x}{H_a}$ for a given value of $x$, and the smaller the variation of $M$ from the desired value 1.

20 Fig. 4 shows the curves of $M$ for different velocities of intake (600, 1000 and 1600 ft. per min.). These curves show that the intake velocity in the mixing valve must be at least 1000 ft. per min. in order to avoid disturbing variations of the mixture due to accidental changes of the gas pressure. Practical experience fully confirms the correctness of this consideration. Velocities of about 1200 ft. per min. must be considered as normal. In order to avoid too intensive a throttling action and consequent reduction of power, the piping and other passages leading to the mixing valve must be proportioned liberally, so that the total drop in pressure is advantageously concentrated within the valve itself.

21 The foregoing calculations are based upon the simplifying assumption that the flow into the mixing chamber takes place at a uniform rate and without loss.

22 For a long time great difficulties were presented by the problem of driving variable speed blowing engines by four-cycle gas engines. In most cases the gas arrives at the mixing valve under a pressure of several inches of water above the atmosphere, while the pressure of the air is somewhat less than atmospheric. When the machine is run slower the suction action of the piston is reduced and the intake velocity of the air decreases more rapidly than that of the gas, until finally it is reduced to such an extent that nothing but gas enters the cylinder. It is self-evident that, even before this point is reached, the engine will choke with gas and will stop. A governor which regulates the quality of the mixture would only favor this suffocation of the engine with gas. In such cases, too, the conditions are improved by intensive throttling at normal speed.

## CYLINDERS

23 Improvements of design as well as improvements in foundry practice have reduced the breakage of cylinders to a point where it is now rather a rare occurrence. Views upon the most suitable shape still differ, however, considerably. While in all other details of large gas engines standard designs have been developed, which serve all purposes, the cylinder designs still vary considerably. The fact that some designers, after careful experiments, have abandoned the split jacket cylinder in favor of the normal one-piece cylinder, while others, also after careful experiments, have gone in exactly the opposite direction, and split not only the jacket, but also the inside cylinder, Fig. 5, seems to show an uncertainty in judging the causes of breakage. These differences in design, however, are caused largely by fixed ideas

of the purchasers whose special wishes are complied with by clever salesmen. The one-piece cylinder is just as strong as the split one. The split cylinder has come principally from the desire to avoid initial stresses in the direction of its axis, which put the inner cylinder under tension on account of the fact that in casting it cools later than the rest of the casting. Besides, by casting the two halves separately, it was attempted to obtain as close-grained a wall as possible for the combustion chamber. Finally, splitting the cylinder has the advantage that customers who consider a cylinder liner the proper construction can be satisfied, inasmuch as the insertion of liners in one-piece cylinders presents some difficulties, though these are not to be considered insurmountable.

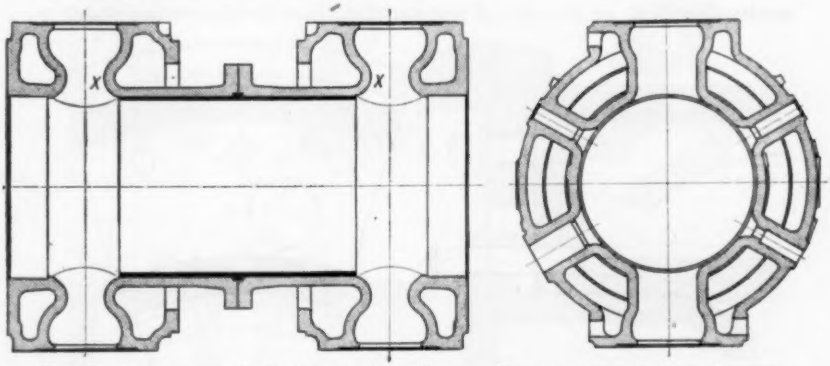


FIG. 5 DESIGN OF SPLIT-JACKET GAS ENGINE CYLINDER WITH SPLIT INSIDE CYLINDER

24 Besides the advantage of simplicity and the absence of the danger of leakage in the highly strained cylinder joint, the one-piece cylinder allows a better transmission of the forces from the cylinder to the frame. The fact that one-piece cylinders frequently broke at the place indicated in Fig. 6, caused an over-estimation of the stresses in the casting. Closer investigation shows that the breaks are caused simply by making the part between cylinder and flange too weak. The wall, besides being too weak originally, is further weakened by the large number of cover studs and the break is further induced by excessive tightening of these studs. This fault can be remedied by properly reinforcing the point of danger and by placing the joint shoulder on the outside. As a matter of fact, breakage on account of strains in the casting or from expansion due to heat, has not occurred in cylinders that were properly reinforced. Improvements in foundry practice have undoubtedly helped to avoid such breakage.

25 While the breaks just discussed have their origin in the strains set up by irregular cooling of the casting or by irregular heating in operation, cracks of an entirely different nature have been observed on the walls of the combustion chamber. These so-called fire cracks always start in a place where the transmission of heat to the cooling water was impeded by some cause or other (see Fig. 7). The cause of these cracks is to be found in the stresses produced by unequal temperatures within the same wall. In the smooth and homogenous wall the differences in temperature of the different strata are small and the temperature of the wall is comparatively low, as long as accumulated scale or similar causes do not offer an obstacle to the transmission of heat. When, however, the capacity of the wall for conducting heat, or the rate of transmitting heat to the cooling water is

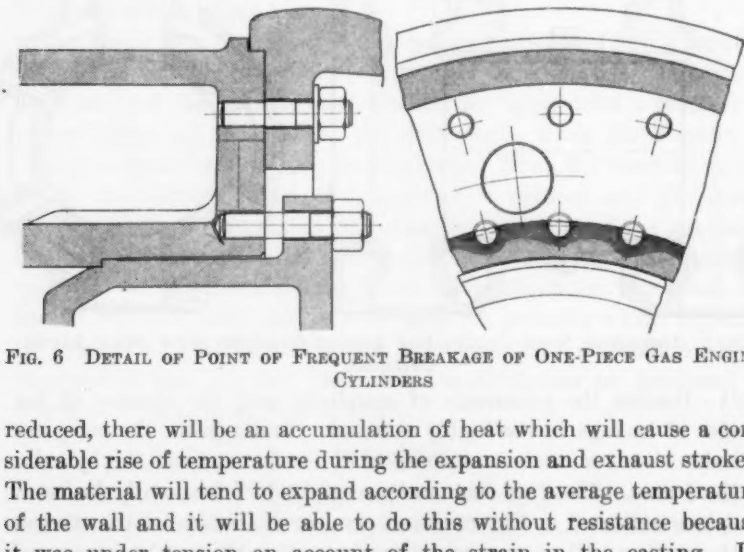


Fig. 6 Detail of Point of Frequent Breakage of One-Piece Gas Engine Cylinders

reduced, there will be an accumulation of heat which will cause a considerable rise of temperature during the expansion and exhaust strokes. The material will tend to expand according to the average temperature of the wall and it will be able to do this without resistance because it was under tension on account of the strain in the casting. In operation, therefore, the strain will be relieved. On account of temperature differences, however, in different strata of the wall, there will be stresses in the wall itself, compression in the hotter zones and tension in the cooler ones.

26 This condition of stress can be compared with that existing in a bar being bent towards the inside of the cylinder. In the hotter layer the expansion being resisted causes compression stresses, and in the colder layers there will be tension stresses. As soon as the cold mixture is admitted during the next suction stroke, the surface of the

wall is cooled intensively. The inside layers of the wall cannot follow rapidly enough to cause the establishment of settled conditions corresponding to this flow of heat. The mean temperature of the wall, and consequently its average expansion can only be affected very slightly during the first moment of internal cooling by the entering cold mixture. The innermost layer will be under strong tension on account of the sudden cooling. This "jumping" of the temperature acts upon the material in much the same manner as sudden flexure from the inside to the outside would. The inside layer of the wall of the combustion chamber, therefore, is exposed to the same kind of stresses as

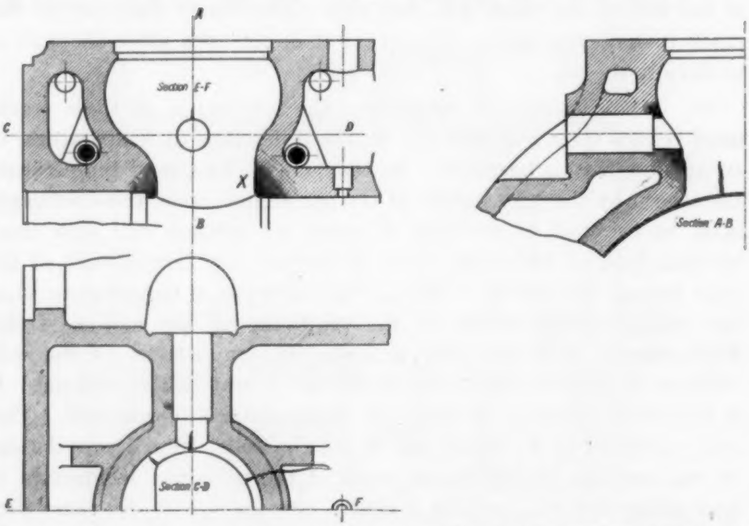



FIG. 7 TYPICAL LOCATIONS OF FIRE CRACKS IN PLACES WHERE COOLING IS IMPEDED

the outside fiber of a bar that is continuously bent in both directions by blows. The stresses which occur under these conditions are about proportional to the difference in temperature, the coefficient of heat expansion, and the modulus of elasticity, if, indeed, one can speak of such in the case of cast-iron.

27 This is an extraordinarily unfavorable case of strain. Cases of strain on account of sudden changes of temperature are not rare, but these conditions are worst in the cylinders of gas engines, on account of the rapid succession of the changes in temperature. A mathematical deduction of these stresses is out of the question on account of the impossibility of obtaining even halfway accurate data on the distri-

bution of temperature at any one moment over the different zones of the wall.

28 If we bear in mind, however, that a difference in temperature of 200 deg. fahr., when the expansion is restrained, corresponds to a strain of about 15,000 lb. per sq. in., we are surprised to find that cracks do not occur more often. The occurrence of a crack, which in the beginning is scarcely 1/8 in. deep, apparently relieves the strain to a certain extent and only the natural tendency of the cast material to continue to break together with the external mechanical forces causes the crack to open further. In most cases drilling and calking at the end of the crack will stop this. The timely discovery of the crack is, however, rather difficult, as it is not open after the wall is uniformly cooled.

29 The first step in combatting the occurrence of these cracks must reduce their real source, i. e., the differences in temperature, to an amount that is harmless. As the sudden changes of temperature are caused by the very nature of the gas engine cycle, these attempts must be confined to avoiding all irregular ignition and slow combustion, both of which are liable to increase the temperature of the cycle beyond the normal as well as the difference in temperature when the sudden change occurs at the beginning of the suction stroke. Furthermore, it is necessary to make the conductivity of the wall uniform in order to avoid accumulations of heat in the material. It is therefore necessary to avoid all accumulations of material. Passages as shown at *X*, Figs. 5 and 7, are also bad. Here accumulations of heat are the natural consequence of the imperfect conduction of heat being directed towards a center, the passage at the same time being very much exposed to the cooling action of the incoming air. Places like these are predestined to suffer from heat cracks.

30 The commonly used globe or onion-shaped passages for inlet and exhaust valves are therefore not suitable, and lately the form shown in Fig. 8 is very properly preferred. Here the valves are brought close to the inner surface of the cylinder.

31 In a much more effective manner than by measures of design, can the heat cracks, and in fact all cracks that occur in gas engine cylinders, be avoided by proper choice of a material. The constant of this material (coefficient of heat expansion multiplied by modulus of elasticity), must be less than is the case with cast iron. The less the expansion from heat, which is the real cause of the strain, and the more elastic the material, the less the strain.

32 Considering the enormous progress which metallurgical science has recorded during the past few years, a solution of this question of material should appear possible, and the more so, as nickel steel alloys have actually been made for accurate rules, in which expansion from heat cannot be detected at all.

33 An investigation of the constant in question (coefficient of heat expansion multiplied by modulus of elasticity) upon which depends, according to the foregoing, the strain of the material resulting from uneven temperature, shows that cast steel is not a suitable material for gas engine cylinders. The larger expansion due to heat as compared with cast-iron and the very much higher modulus of

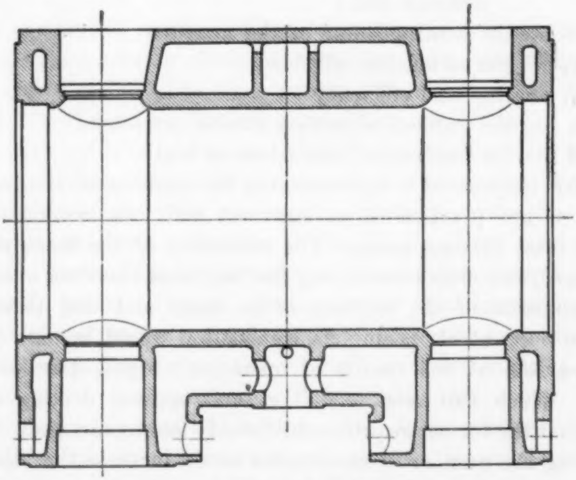


Fig. 8 Recent Design of One-Piece Gas Engine Cylinder that Avoids Globe-Shaped Passages to Valves

elasticity causes increased stresses, while the strength of cast steel is not proportionately greater. Experience confirms this conclusion.

## ON INCREASING THE OUTPUT OF LARGE GAS ENGINES

34 The principal task upon which the gas engineer is laboring incessantly, besides that of increasing the reliability of the machine, is that of making it cheaper, partly by simplifying its construction, partly by increasing its output. At present it can hardly be imagined that the gas engine can be made cheaper by making its construction simpler, now that the complicated valve gears have disappeared. Reduction of weight is not to be recommended. Until recently experience has demanded a continuous increase of engine weights. Economy

in material would reduce reliability and length of life of the machine. The mechanism has been developed into standard designs, so that in this detail there is nothing to be saved.

35 Somewhat better prospects seem to be opened by the attempt to reduce the cost by increasing the output of the machine of a given size. Let us consider the equation for the power of the machine

$$W = \frac{1}{A} H \, V \, \gamma \eta_v \, \eta_t$$

in which

$V$ = the swept volume (the factor which determines the absolute cost)

$H$ = the heating value of the mixture

$\eta_v$ = the volumetric efficiency

$\eta_t$ = the total efficiency

$\gamma$ = the number of suction strokes per second

$A$ = the mechanical equivalent of heat

36 An increase of $\gamma$ by increasing the number of revolutions per minute, is not practicable, at least not with less combustible gases such as blast furnace gases. The reliability of the machines would be lessened, not even considering the fact that the time available for the combustion of the mixture is too short and that therefore the losses on account of incomplete combustion would become too large. A comparison of the results of operation of gas operated blowing engines, which run relatively slow, and engines driving dynamos, proves that the former require considerably less repairs than the latter. Increasing the number of revolutions would increase the idle time on account of repairs, and a gain in output can hardly be expected.

37 The heating value of the mixture $H$, is also determined by the minimum excess of air necessary for complete combustion, and there remain only the two factors of volumetric and total efficiency. It is conceivable that the total efficiency might be increased by increasing the thermal efficiency of the process. This, however, is dependent upon increase of the pressure and for this reason such a possibility must be eliminated. The volumetric efficiency can be increased in two ways; by cooling the charge and by increasing the pressure of the charge. The first method was proposed some time ago. It is not being adopted because in most cases the cooling water is warmer than the atmosphere and the incoming gas. Besides, a large gain cannot be expected, as decreasing the temperature of the charge 10 deg. fahr. could only increase the output about 2 per cent.

38 The other method for increasing the output, which lately has attracted renewed interest, consists in increasing the pressure of the charge and simultaneous scavenging of the exhaust gases. In this manner a given suction volume of an engine can accommodate a larger weight of charge and greater output of power can be attained.

39 The original purpose of this method, which has been known in England for many years, was the avoidance of pre-ignition by scavenging the cylinder of what remained of the hot exhaust gases. It is the competition into which the gas engine has lately had to enter with the steam turbine, that has brought to the front the possibility of increased power output and consequently lower cost of the machine per unit of power.

40 The simplest method of realizing the scheme of scavenging and charging consists in closing the exhaust valve late and opening the inlet valve early. The time during which both valves are open is used for scavenging. In order surely to avoid loss of gas, the charging with gas is only commenced after the exhaust valve is closed. At the end of the suction stroke the mixing chamber must be cleansed of combustible mixture by scavenging (rinsing) with pure air, as otherwise backfires would occur at the beginning of the next scavenging period. Gas and air are brought to the machine under a pressure of about 3 lb. per sq. in. gage. On account of this increased pressure the weight of the charge is about 20 per cent greater than that of the normal method. Scavenging the combustion chamber also furnishes further space for fresh mixture. Experiments have shown a mean effective pressure of about 100 lb. per sq. in. In continuous operation, in so far as we can speak of such in experiments, mean effective pressures of 85 lb. per sq. in. can be reached.

41 Undoubtedly such results present much that is attractive, especially as the output can still be increased by further increasing the pressure, and as this would give to the gas engine the capacity for overloads which heretofore it did not possess in the same sense as steam engines do. Experiments have also proved that these overloads can be had without an increase of the maximum pressure. This is done by reducing the compression.

42 In spite of the lower compression the heat consumption per unit of power was not increased. It seems that this result is explained first by better combustion of the charge which is not contaminated by the remains of the exhaust gases, and secondly by a decreased percentage of loss in the cooling water. The predominating influence can only be found by means of carefully measured heat balances which at present

are not available. The increased temperature of the cylinder walls, which has been found thermo-electrically, has caused objections against continued operation with scavenging and forced charge. A certain justification cannot be denied to these objections. The *decrease* of continuous loads is the main factor that has contributed to overcome the difficulties of operation. To endanger regular operation by increasing the mean effective pressure must appear to be a dangerous experiment in just such cases where the equipment is insufficient and an increase is highly desirable. The disposition on the part of operating engineers to await developments, is easy to understand. The present satisfactory operation, which is the result of years of experience, is making these engineers justly conservative.

43 The danger is to be found in the fact that the increase of output on one hand is offset by a decrease on the other hand on account of the more frequent shutdowns for repairs. Besides these questions, which can only be solved by several years of experience, there are other difficulties in the constructive realization of this idea of increased output which have to be solved before it is proved to be practicable in sustained operation. In one word, there are many nuts to be cracked, before we can think of a practicable increase of the power output of four-cycle engines by means of scavenging and forced charge.

## IMPROVING GAS ENGINE HEAT ECONOMY

44 The endless desire of gas engine builders to improve the economy of the gas engine, has lately brought to the front interest in the utilization of waste heat. According to experience, a blast furnace gas engine dynamo consumes in continuous service on the average about 16,000 B.t.u. per kw-hr. Of this amount of heat only 3412 B.t.u. are converted into electrical energy. Of the balance, 12,588 B.t.u., only the mechanical, electrical, and radiation losses have to be regarded as unredeemable, as well as the energy contained in the unburned gases. The heat in the cooling water, about 4800 B.t.u., and that in the exhaust, about 5200 B.t.u. per kw-hr., is, however, available for a more or less perfect further utilization. The high temperature of the exhaust, about 880 deg. fahr., measured at the exhaust flange, makes possible its immediate use for the generation of steam in boilers, which have to be placed as closely as possible to the exhaust chamber. As a matter of fact, it is possible to obtain about 2 lb. of high-pressure steam per kw-hr. of the gas engine.

45 The utilization of the heat in the cooling water is not possible as directly as this. The further use of this heat for power purposes means the generation of steam and this requires a considerably higher temperature of the cooling water than that customary today in large gas engines. Semmler suggested as long ago as ten years, that gas engine cylinders be cooled with water hotter than 212 deg. fahr., and that this water be put under pressure in order to avoid the generation of steam in the jacket. Evaporation commences only outside of the jacket, when the superheated water is conducted through a throttle valve into a steam drum. It may be easily understood that this "hot cooling" scheme has been regarded rather skeptically for a long time. About a year ago it was decided at the Rombacher Huetten Werke to try Semmler's cooling method on an 800-kw. tandem engine. The results in operation, which are available today, are entirely favorable. The hot jacket not only did not hurt the cylinder, but it caused apparently a softer and more quiet operation of the machine.

46 The use of distilled water in a closed cycle like Semmler's, the considerably higher temperature of the outer cylinder wall, which causes nearly equal average temperatures of the inner and outer cylinder walls, are surely advantages which, if the favorable experiences last, merit more consideration than the recovery of waste heat. The amount produced is in the case of the engine mentioned, about 0.5 kg. of steam of low pressure per kw-hr. There is no doubt that this figure can be very much improved by designing the cylinders more suitably for this purpose, by careful insulation and by connecting the cooling water from the piston with this system, which heretofore has not been done.

47 The results achieved in recovery of waste heat will be of different practical value for different types of blast furnace plants. Where there is a steam plant in addition to gas plants, it would be advantageous to shut down some of the boilers and substitute steam from the waste heat boilers by connecting them to the steam mains, because this steam can be produced almost without cost for attendance and for fuel, and therefore, as cheaply as is possible.

### GAS ENGINE VS. STEAM TURBINE

48 In conclusion it may be well to touch in a few words, upon the competition between the gas engine and the steam turbine, which lately has become very keen in Germany as well as in the United States. The steam turbine has doubtless made much progress

during the last few years. In its perfected form in which we know it today, it is a thoroughly reliable machine. Its overload capacity, with nearly constant steam consumption per power unit, is limited by its accessories, as boiler and condenser. In addition, there is its principal advantage of lower first cost as compared with the gas engine. Its disadvantage is the higher consumption of heat per unit of power. The gas engine produces approximately twice the amount of energy from a given amount of heat, and is therefore, superior as a fuel saver from the economic point of view. To the business man, however, the question of lower first cost often appeals more strongly. The favorable balance sheet of the current year pleases him better than the quiet satisfaction in the knowledge that he has saved valuable treasures of fuel for his great-grandchildren. But even from this standpoint the gas engine will be victorious over the steam turbine in cases where the available waste gases are not so abundant that their combustion under steam boilers would suffice to satisfy the existing requirements for electrical energy. Whether and to what extent these conditions will be changed by the new method of surface combustion, is a matter which the future will show.

## DISCUSSION

H. J. K. Freyn. Professor Langer devotes a considerable portion of his paper to the question of regulation of large gas engines and he points out in a very able and clear manner that of the two prevalent systems of regulation of gas engines, namely: by stratification of the mixture and constant compression versus throttling of gas and air and variable compression, the latter method has proved its superiority in every instance and from every point of view.

I can but heartily endorse every statement made by Professor Langer and while in my earlier career I favored the stratification method, I changed my mind many years ago as a result of practical experience with engines regulated on both systems.

Abroad, the leading gas engine manufacturers have for a considerable length of time adhered to the stratification principle, although choosing the more rational form of attempting to obtain stratification by arranging their valve gear in such a manner that a certain quantity of pure air always *followed* the mixture.

In this system, therefore, only one chance for diffusion between gas mixture and pure air exists, with the result that operation at fractional load was reasonably satisfactory because the danger of

formation of a bad mixture during the suction and compression strokes was not as great as it is in the system where a variable quantity of mixture is "sandwiched" between two layers of pure air, causing diffusion in two planes, as it were, on either side of the mixture.

It has been my experience that gas engines operating on the so-called stratification principle, while giving excellent results from full load to approximately half load, are not capable of maintaining regular ignition upon all piston faces as soon as the load drops below approximately 50 per cent of the rated capacity.

The influence of this phenomenon upon regularity of operation, especially if alternating-current generators driven by such engines have to be operated in parallel, is very marked and it will be found that such power plants usually show excessive cross currents and "swinging on the line" at light load. This is not the case with engines regulated on the principle of variable compression and constant mixture, and especially with those where gas and air are throttled during the whole suction stroke.

Professor Langer has very ably and comprehensively proved along theoretical lines why this should be so, and he points out especially what provision must be made in the governor gearing to bring about the desired results, not only at full load but also at fractional load.

The point of "drowning" of gas engines in gas, raised by the author, is very well taken. Gas blowing engines which have to be operated at times at reduced speed are particularly susceptible to drowning. Gas blowing engines regulated on the principle of constant mixture and variable compression are better suited for slow speed operation, as was proven in many instances in Europe where large gas blowing engines could be operated for one hour at a speed of 19 r.p.m. with full pressure of 15 lb. on the blowing cylinder.

Without a single exception, German gas engine manufacturers several years ago abandoned the stratification method of regulation and a large majority have adopted the so-called "combination system," making use of the simplest method imaginable for throttling of air and gas, viz., by the application of so-called butterfly valves as throttling and regulating organs.

These engines are regulated in the following manner. From full load down, the governor acts first only upon the gas dampers with decreasing load, while the air dampers remain in their original position. Below about 60 to 70 per cent of rated load, the governor begins, through a toggle motion, to act upon the air butterflies as

well. The toggle effect is so arranged that with further decreasing load, the air dampers act more quickly than the gas dampers, so that below a certain fractional load a perfectly constant mixture is admitted.

It will be seen that this combination method of regulation combines the advantages of both stratification and constant mixture regulating methods; at the heavier loads, practically constant compression is obtained, while at the lighter loads governing is performed on the constant mixture principle.

With reference to the subject of gas cylinders, I believe that Professor Langer's excellent exposition of the relative merits of cast iron and cast steel and of the one-piece vs. the split gas cylinder deserves the careful attention of both gas engine builders and users. Professor Langer's discussion of this question is very timely, because in this country the battle between the advocates of the one and those in favor of the other is still waging.

I have had an opportunity of studying these questions in detail during the last few years and I have come to the conclusion that Professor Langer is unquestionably right in his preference for one-piece cast-iron gas cylinders fitted with hard cast-iron liners. Earlier difficulties with cracking of gas cylinders have naturally led manufacturers to look for a building material which could stand up better under the strains and stresses imposed by mechanical forces and temperature variations. In Europe cast-steel cylinders were tried several years ago with the result that breaks occurred after a much shorter time of operation than when cast-iron cylinders were used. I believe that a similar experience was had in this country.

Cast-iron cylinders in one piece can be made so perfect today that they will stand up very well. At the gas engine power plant at Gary, 140 one-piece cast-iron gas cylinders are in operation, 68 of which have been in service nearly five years. Of all these cylinders only one had to be actually replaced by a new one, although at the time of installation of these particular engines, the art had not progressed as far as it has today. It is true that a number of these cylinders show fire cracks and breaks in the counterbore, but repairs have been made and the cylinders are still in perfectly satisfactory operation.

One of the most important points to which by far too little attention is paid, especially in this country, is the question of the use of suitable cooling water. Very interesting experiments were made abroad to determine the effect of the accumulation of scale on the

cylinder walls and it was found that the average temperature of these walls increased at an amazing rate with seemingly unimportant deposits of scale. I know of at least one installation where thermometers are inserted permanently in the inner cylinder walls reaching to within 1/4 in. of the cylinder bore. The operator is supposed to watch these thermometers which, after the cylinder jackets have been thoroughly cleaned with wire brushes and weak acid solutions, show a gradual rise of temperature while operation continues. As soon as the thermometers show a temperature of 125 deg. cent. the engine is shut down and the cylinder walls are again carefully scrubbed and cleaned. In this particular instance the cracking and breaking of gas cylinders which had become a nuisance on account of its frequency, was entirely stopped.

Professor Langer has elaborated considerably on the question of the increase of output of large gas engines and improvements in gas engine economy. I have studied the latest attempts of European gas engine manufacturers to obtain an overload capacity of four-cycle gas engines by using the so-called scavenging and surcharging method. I have seen several installations abroad equipped with this system and I am familiar with tests made by Ehrhardt & Sehmer on engines furnished by them. They show that in spite of a material increase in mean effective pressure amounting to 25 to 35 per cent, no increase occurs in the initial pressure nor in the average temperature of the gas cylinders and other gas engine parts exposed to the high temperatures of combustion. This firm is prepared to give guarantees regarding the heat consumption of engines equipped with the scavenging and surcharging system which are not any lower than those usually given for ordinary four-cycle engines of equal capacity.

I cannot agree with Professor Langer's statement that engines operated on this principle are subject to a greater wear and tear than ordinary gas engines; the stresses and strains which the running gear and other engine parts sustain are due primarily to the initial pressure and not to the mean effective pressure. If we consider the main bearings, for instance, it will easily be seen that such a bearing will give no trouble as long as a film of oil can be maintained between bearing shell and shaft. The maintenance of such a film depends upon the amount of pressure per square inch and not upon its duration; with lower initial pressure and higher mean effective pressure, therefore, these main bearings must give at least as good satisfaction as they are giving in ordinary four-cycle engines. As a matter of

fact, the dimensions of pins, bearings etc., of scavenged and surcharged engines have not been increased beyond those customary in ordinary non-scavenged engines.

The question of utilization of the exhaust heat of gas engines has had practically no attention in this country, and yet it is such an excellent and cheap means of increasing the usefulness of the gas engine and the financial returns from its application. I wish to refer in this connection to one of the best papers on the subject, which was read by Leon Greiner before the Liege Engineering Society about a year ago. Practical results are given which were obtained with an installation of such boilers in connection with blast furnace gas engines at the John Cockerill Works of Seraing, Belgium.

Referring to the competition between gas engines and steam turbines, I wish to call attention to my paper read before the American Iron and Steel Institute's meeting of last May, which is devoted exclusively to this subject and which proves with the aid of cost figures obtained in actual, commercial operation of large gas engine power stations in this country that gas engines operated on so-called "waste" industrial gases—an unfortunate misnomer which should be abolished as quickly as possible—are superior in commercial economy to any other known method of producing power.

F. Z. Nedden said that the present status of the large gas engine, both in Europe and in this country, is perhaps more considerably influenced by the introduction of surface-combustion boilers than implied in the paper. In this country we have very largely the use of natural gas, especially in the Pittsburgh district, which is very suitable for being used under Schnabel-Bone surface combustion boilers. One of these installed at the Shinningroove Iron Works, in England, is claimed to have given an efficiency of nearly 90 per cent, as compared with the efficiency of the ordinary boiler, which ranges from 65 to 70 per cent. That would mean that with the surface-combustion boiler the gases available in the Pittsburgh district would be able to produce one-third more power than at present. In other words, the cost of energy would in that district be reduced by about 20 per cent.

A great deal is said about the advantages of the Schnabel-Bone boiler which are naturally claimed by those interested in its introduction. Professor Langer's experiences with it would be of interest, if he would give some account of them.

To make his questions more specific, he said he would ask Professor Langer if it was true that the efficiency of the Schnabel-Bone boilers actually exceeded that of the ordinary boiler by about 25 per cent, and if that boiler was reliable in service[1]; also if the utilization of the heat of waste gases in the gas engine was possible by the Schnabel-Bone boiler?

R. H. Fernald. At a central plant in this country, built to utilize low-grade fuels at the mines, I ventured to criticise the fact that a steam plant had been constructed instead of a gas plant. The vice-president of the company said that he had wanted to put in a gas plant, but was forced to put in steam as he found no large gas producer units that would be available and satisfactory for an installation of the size contemplated. In other words, it seems to be a question of the size of the unit which is deferring the installation of plants of 100,000 to 250,000 h.p. directly at the mines. I would, therefore, like to hear from Professor Langer his opinion regarding the possible size of the large gas engine of the near future. The question, of course, arises as to what is a large gas engine. In 1900 a 500-h.p. engine was considered large. At the present time we have 5000-h.p. units. What is to be the large gas engine 10 years from now? The reciprocating steam engine was slow in developing. The steam turbine developed more rapidly than the reciprocating steam engine.

All are familiar with the relic in the yard of the General Electric Company at Schenectady—a 5000-h.p. unit which was one of the early turbines of this country. This was followed by an 8000-h.p. turbine and later by a 14,000-h.p. unit, and I understand that an order has recently been placed for a single unit of 30,000 or 35,000-kw. capacity. This steam turbine development has been very rapid.

If we are to install large central plants at the mines and develop electric current for long distance transmission we must have large gas engine units if producer gas is to compete with steam. A gas producer of 3000 to 4000-h.p. capacity in a single shell has recently been installed. The construction of the plant is so simple that the cost of manufacture should be low. If this producer meets the demand and proves to be a commercial proposition, there is a possibility of developing units of not less than 10,000 h.p. in a single shell (since these comments were made I understand that a unit of 75 tons

[1]This question is very fully answered by G. Neumann's article, an extract of which appeared in The Journal, Am.Soc.M.E., January 1914, Foreign Review Section, p. 09, etc.

capacity per 24 hours has been ordered), but gas engine units of a size to compete with large steam turbines are slow in developing. This central station at the mines is not pure theory. The few installations that have been made at the mines have attracted the attention of engineers and I understand that one of the large coal companies of this country is at the present time considering the installation of such a plant for the utilization of mine refuse. I was told recently that representatives of this company are in Europe seeking information relating to large by-product gas plants. I have also been told that the producer-gas interests are also alive to the situation and have sent representatives abroad within the past few weeks to study the question of large units. These large units seem essential if we are to utilize our fuel resources to the best advantage. I trust that Professor Langer can give us some definite information regarding the future of the large gas engine unit.

F. S. Giller. With reference to the point brought up about having gas-driven electric plants at the coal mines, I would like to ask whether gas engine sets, in their largest sizes, could compete with steam turbine sets, in their largest sizes, in the matter of cost of production and convenience of operation.

It is not easy to understand just what is meant by the author's statement that the gas engine is vastly superior as a fuel saver from the economic point of view. A concern handling a large power production undertaking does not consider either cost of fuel or first cost, except in their relations to the total cost of production. Undoubtedly the cost of fuel is a very important item in this total cost, but I believe that practically all of the other items, such as depreciation, repairs, rents, taxes, insurance, interest on investments, etc., are higher for gas plants than for steam plants, and that together they usually constitute a greater argument against the gas engine sets than the high heat efficiency does for them. I have lately visited many large steam-driven and water-driven electric generating plants, and also the large gas-driven plant at Gary. At this last place, I could not help being struck by the great size of the generating sets in comparison with the sizes of corresponding turbine sets, the enormous foundations and buildings required by them, and the obvious need of a large staff of men to keep them in proper repair and operation. Undoubtedly the sets would be smaller if they were using richer gas, but I doubt whether, at the present time, the best gas engine set, working under its best conditions, can compete with the best steam

turbine set, working under its best conditions. With regard to the future, the chance would seem to favor the turbine more and more, for the efficiency of the turbine is improving rapidly and continuously, whereas that of the gas engine is not so marked. The author speaks of the competition between the gas engine and the steam turbine, which lately has become very keen in Germany as well as in the United States. I am under the impression that the competition is becoming less keen and that the arguments are more and more in favor of the turbine. Several years ago, the Mond people in England were producing and distributing power gas on a large scale, but I believe their installations do not increase much, and that the progress made by gas plants generally during the last few years has not been anything like so marked as that made by the turbine, except that gas plants have been utilized in steel mills and other places where gas is to be had under unusually favorable conditions.

A comparison of the theoretical costs of production of electric energy by a large gas-driven plant and by a steam-driven plant of the same total capacity, assuming both of them to be situated at the coal pit and burning all the coal brought up from it, would be very interesting. I would like to ask whether such costs have ever been prepared and if not, whether the author considers that they would argue in favor of the gas-driven sets.

F. Z. Nedden, in reply to a question, referred to the Humphrey pump recently installed in Chingford near London. The problem of using the Humphrey pump as a generator of electricity has been taken up by the Siemens-Schuckert Works of Berlin. They are actively engaged in tests with the object of generating electricity by means of a water turbine driven by water raised by a Humphrey pump. The water after passing the turbine circulates through the pump again. So far nothing has been published as to the economy of the system. Such a plant would seem to be the proper spare for a low fall water turbine during periods of shortage in water. Instead of using a separate spare generator set driven by a steam or internal-combustion engine, the water would simply have to be pumped, by a Humphrey pump, from the tail-water back to the headwater and flow through the water turbine again, thereby saving expense for a spare generator and simplifying service.

The Author. Referring to the discussion by Mr. Freyn, while regulation by throttling the gas only will produce a smoothly running engine, I prefer to throttle air and gas simultaneously.

In reference to cylinders, cracked cylinders which have been calked have lasted a good many years and very many of them are still in use. It does not weaken or damage the cylinder if cracks are noticed soon and calked. The crack gives some relief to the lamina which is exposed to the highest temperatures and strains.

With regard to the objections in connection with the surcharging of the machine mentioned in the paper, I do not want to imply that the strains are due to the greater mechanical forces acting. There is no question that by reducing the compression pressure it is possible to work with the same maximum explosion pressure, at the same time having from 20 to 25 per cent higher mean effective pressure. The strains referred to are not caused by mechanical forces but by heating. In this matter I cannot draw on my own experience, but I have heard that the temperatures of the cylinders are increasing. This would mean an increase of strain in the cylinder walls due to the surcharging of the engines.

In reply to Mr. Nedden, I have no special information on the surface-combustion boilers. Generally it is said that the cylindrical surfaces on which the combustion takes place are filled with dust in a short time and put out of service. This, I think, is not a serious inference, since it should be possible to produce combustion cylinders at such a cheap price that it will be practicable to "scrap" them when necessary.

Professor Fernald asked about the size of gas engines. When we speak of large gas engines it means engines of 1000 h.p. and upwards. Double-acting engines of less than about 35 in. or 36 in. diameter of cylinder are not built at all, at least in Germany. The limit of the size, according to our experience there, is about 55-in. bore, by 6-ft. stroke. Tandem engines of this sort would carry a load of 3500 h.p., which means a twin unit of 6000 or 7000 h.p., about the maximum limit. This limit is not determined by shop practice or by designing, but by the impossibility of shipping by rail larger sizes of frames, cylinders and tie pieces.

I agree with Professor Fernald with regard to the production of electric power at the mines. Power can be produced at the mines for generating electricity and the current transmitted at 100,000 or 150,000 volts, which voltages I understand are being used in this country. Whether these big central stations will use gas engines or steam turbines is very hard to say; the question will probably be decided to a large extent by financial considerations.

While big units with piston engines are limited to 6000 or 7000 h.p., no limitations are given for the gas turbine, and this problem certainly represents the next advance to be made by the mechanical engineer.

In reference to waste heat boilers (in reply to a question), I cannot give the data on the necessary heating surface of these boilers. But it is not advisable to use too much of heating surface, for it is necessary to discharge waste gases with temperatures of 200 or 220 deg., in order to prevent the condensation of the steam vapor in the exhaust gases and corrosion of the tubes, as sulphur is found in any of the industrial gases.

[illegible]

No. 1420

# THE FIRE HAZARD IN TURBO-GENERATORS

By G. S. Lawler,[1] Boston, Mass.

Non-Member

The chances of electric generators of the older types being seriously injured by fire in the event of some part of the insulation failing is slight. Occasionally arcing will ignite the insulation at some point, but it is seldom that the fire will spread much before it is extinguished. This freedom from fire damage is due principally to the comparatively low speeds, the accessibility of the combustible insulation, and the fact that the machines being of large mass per unit capacity, the insulation is considerably distributed.

2 This condition of practical freedom from fire is reversed in the case of generators of the turbo type, for when a short circuit occurs in one of them there is a great chance that the insulation will be ignited and the machine be badly damaged; in fact such damage has occurred in a number of instances.

3 The chief causes of the increased hazard in the more modern type generators are as follows:

*a* The volume occupied by this type of machine is very much less for the same capacity than that of the older types of generators, so that the combustible insulation is more concentrated and, therefore, much of it is exposed, even to a slight arc or fire. The covering on the conductors depends greatly for its insulating qualities on the presence of oils or gums of a highly combustible nature. The amount of this combustible insulation on the higher voltage generators is naturally greater than in the low voltage machines.

Owing to turbo-generators having only a few poles the end connections between slots form a large proportion of the total length of conductors; in fact in some designs

[1] Elec. Engr., Inspection Dept., Assoc. Mut. Fire Ins. Cos., 31 Milk St.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

approximately one-half of the coils are outside of the slots. These end connections, one-half being on one side of the machine and one-half on the other, are exposed to fire, and as with a pile of loosely laid sticks, fire will rapidly extend from the insulation on one coil to that on the others.

*b* Owing to these generators being of exceedingly large capacity in many instances (one of 30,000 kva. capacity being now in process of construction), an enormous amount of energy is involved in a short circuit, especially at the instant the short occurs and as the arc is confined in the limited space with the combustible insulation, it would seem impossible for the insulation to escape being set on fire at many points simultaneously.

*c* The machines are cooled by forcing large quantities of air through the spaces between the conductors. The large and constantly renewed supply of oxygen will hasten combustion when it is once started.

The air is given somewhat of a rotary motion by the rapidly revolving rotor which has the ventilating vanes on it and consequently fire when started will be quickly swept around the exposed insulation.

*d* The generators are totally encased with the exception of the air inlets and outlets and even these in some designs are under the machines. This construction prevents access to a fire and much valuable time will necessarily be consumed before extinguishing agents can be used effectively. When the field current is cut off, as is necessary in case of short circuit, the only means of bringing the rotor quickly to rest is lost and it will continue to run for a long time after the steam has been shut off. Some machines will run for over an hour. This continued rotation is not conducive to the quick extinguishing of fire, especially when the ventilating vanes are mounted on the rotor.

4 In addition to the possible causes of arcing existing in the case of the older types of generators, the turbo-generator is subject to momentary large current rush at instant of short circuit, even if the short is external to the machine itself, unless means are taken to keep the current within safe limits. The heavy rush of current

causes mechanical stresses in the conductors, which in some cases are severe enough to distort the conductors, especially where outside the slots, and to injure the insulating covering, resulting in a short circuit within the generator itself. In some designs the internal reactance of the machines will permit of the momentary current rush amounting to 40, or possibly more, times the normal full-load current of the machines.

5 The possibility of the conductors being distorted has been reduced in some cases by designing generators with sufficient internal reactance, or by providing external reactance such that the current at the moment of short circuit will not be great enough to damage the generators. Attention has also been given to supporting the stator end connections to prevent their distortion. These means have undoubtedly greatly increased the safety of the turbo-type of generator from possibilities of internal short circuit, but in no way tend to prevent a fire resulting should an arc occur.

6 A short circuit in the rotor will probably not result in a severe fire unless under exceptional conditions. This is also true if the short circuit occurs inside of a stator slot. A short circuit involving a stator coil, however, is more apt to occur at the end of the slot where the conductors are exposed.

7 As asbestos is now used largely for insulating the rotor windings and as these windings are well protected, it is probable that only in cases of severe fire in a machine will the rotor windings be damaged to any extent.

8 While the generators may be free from fires during the earlier portions of their life owing to the proper use of reactances which prevent external troubles seriously affecting the machines, as they get older the ordinary causes of breakdown of insulation are liable to occur and fires result. Probably in most cases generators will not be discarded until some trouble, usually in the nature of a short circuit, has occurred at least once in each, so that it is reasonable to expect that unless further preventative means are taken, turbo-generators stand a good chance of serious damage by fire at some time during their life. Although many fires have occurred, probably most of them have happened during the generator development stage. Generators of the turbo type are of such recent production that none of them has yet reached a life which could be considered old and, therefore, the troubles which can be expected near the end of their life by fire have still to come.

9 Undoubtedly the manufacturing companies have given serious thought to the matter of the reduction of the fire hazard in turbo-generators and have employed all means practical at the present time to this end, but there is still very much to be desired. The following several means if taken together would seem to minimize the chances of a serious fire:

*a* If a suitable material could be found, a non-combustible outer covering could be placed over the insulation on the stator end connections. This would greatly delay the spread of fire and even if no other protective means were taken, would undoubtedly prevent much serious damage. Where fire extinguishers were used the covering would at least hold back the fire until they could be brought into play. At present no material suitable for such a covering appears to be available.

*b* If a non-combustible outer covering should be put on, its advantages would be partially lost in time unless the cooling air were freed of the dirt and oily vapor liable to be in it. This could be done by filtering, as has already been advocated several times.

*c* Means could be provided for cutting off the air supply in case of fire in generators by placing dampers in the inlet ducts designed so as to be normally held open by fusible links. The links could be placed so that they would be quickly fused by the heat and allow the dampers to close automatically. By reducing the oxygen supply to that entering by leakage the action of the fire would be slow.

*d* Arrangements could be provided for the quick introduction of carbon dioxide gas into the machines. The carbon dioxide could be kept in liquid form and piped through valves, expansion tanks, etc., to the generators. The valves could be arranged to be opened by the closing of the air inlet dampers so that the gas would be automatically introduced into the generators. This gas would be very effective in extinguishing fires inside the machines after the air supply had been cut off.

10 The employment of some efficient method of reducing the fire hazard in generators of the turbo type either along the lines mentioned or in some other way is important. The value of these generators is great and the damage by fire may amount to a con-

siderable proportion of the first cost. It is probable that the damage is more liable to occur towards the end of the life of the generators, but even then the loss may be large, both directly and indirectly. The large central stations have reserve units so that the increased damage due to fire in one of their generators would probably not affect the continuity of service, but the increased time necessary for repairs may be long and during this time the reserve capacity will be weakened. In the case of industrial plants the longer time needed for repairs might be serious. Many manufacturing concerns who generate their own current depend on only one unit and, therefore, their whole production, or a large part of it, would be affected.

## DISCUSSION

B. G. LAMME[1] (written). Having been connected with the turbogenerator business since its beginning in this country, I am naturally much interested in the conclusions drawn by Mr. Lawler. In general, his statements can be accepted, and about the only criticism which can be made is that he does not go far enough in some of his explanations and recommendations. The following may therefore be considered as a supplement rather than a criticism of Mr. Lawler's paper.

Two main facts stand out in connection with fire troubles in turbogenerators: namely, that the percentage of fires is greater in high capacity machines than in those of moderate or low capacity; and that fires are relatively much more frequent in high voltage machines than in those of low voltage.

As small capacity machines have been in use for a much longer period than those of large capacity, the latter being of comparatively recent date, it would appear that if the age of the unit, or of the insulation, were of first importance in causing fires, there would be a preponderance of fires among the older and smaller units. But such does not appear to be the case, and apparently a principal cause must be looked for elsewhere. Presumably this lies in the greater energy liberated in the case of a short circuit, together with the vastly greater ventilation in the larger units.

If a short circuit occurs in a large unit, the electromotive force producing such short circuit cannot be reduced instantly by cutting off the excitation from the field, for it takes an appreciable time for the field magnetism to die down. This is especially the case in the large, solid rotors, which are now extensively used in such machines.

[1]Chief Engineer, Westinghouse Elec. & Mfg. Co., Pittsburgh, Pa.

Therefore, if a short circuit, resulting in an arc, is established, the energy expended may be enormous, as Mr. Lawler has pointed out, and, even if the field excitation is very quickly cut off, the expenditure of energy may still last for a period sufficient to establish a fire in the end windings. Apparently there is heat enough in such short circuits to "gasify" the insulating varnishes in the neighborhood of the arc, and if this condition is once started, it seems to spread rapidly.

A series of tests was made under my direction to determine whether an asbestos taping over the outside of the end windings would so delay the combustion that the fire might be extinguished without doing any great damage. However, these tests showed that when an arc is once started, sufficient to gasify the varnishes, the asbestos-covered coils burned about as quickly as the others. In fact, there was no encouragement whatever in our tests.

This does not mean, however, that some more effective protective covering may not be developed at some future time. Attention is called to the fact, however, that present available materials, such as asbestos tape, might ultimately represent more of a danger factor than a protection, as has already been referred to by Mr. Lawler. Some very costly experiences have been encountered by electrical manufacturers in the use of asbestos tape as an outside protecting covering in other than turbo-generator apparatus.

As to the second fact stated, that a far greater percentage of fires occurs in high voltage machines, this needs no particular amplification. While a low voltage machine might give as much energy on short circuit as the high voltage machine, yet such short circuits are less liable to happen inside the machine. Moreover, the low voltage machine has relatively less insulation on its end windings and, therefore, there is less tendency to formation of combustible gases. Moreover, the majority of large capacity machines are wound for relatively high voltage, and, as stated before, the larger percentage of the fires occurs in the larger units. Thus the questions of capacity and voltage are so related that it is difficult to say whether either one is more responsible than the other for fire damage.

As to the effect of age on the fire risk, experience has not yet shown definitely that comparatively old machines are liable to suffer from a fire to any greater extent than new ones. This may be due to the fact that old, well baked and well dried insulations do not seem to be as readily combustible as new "green" insulations. The latter will

probably generate gases more quickly than the thoroughly seasoned material. Therefore, as far as the insulating material itself is concerned, age should probably make the conditions no worse. However, accumulations of oil, dust and other foreign materials on the end windings, should make the conditions somewhat worse.

As to fire protection, Mr. Lawler's suggestions are to the point, and the remedies which he proposes should be effective to a certain extent. However, a number of things should be taken into consideration in this matter. He suggests, for instance, automatically cutting off the air supply by closing the inlet, and thus allowing only the leakage air to pass through the machine. This remedy should be fairly effective if the leakage can be made small enough. For instance, a 20,000-kw. turbo generator unit may have 50,000 to 75,000 cu. ft. of air per minute blown through it, in normal operation; this is vastly more than required to maintain combustion if a fire is once started. Even if the main inlet pipe is very completely closed, there might be a leakage of possibly 1 per cent through joints in the inlet valves, in the end housings of the machine, and at other points. This would mean 500 to 750 cu. ft. of air per minute through the machine, which would probably be ample to maintain a very damaging fire.

Furthermore, with even a relatively slight percentage of the normal quantity of air leaking into the machine, it is obvious that a small amount of carbonic dioxide gas would have but little effect. Therefore, the gas would have to be supplied either in great volume, or the air leakage must be made exceedingly small, in order that the gas may become effective. The problem is further complicated by the fact that, in some designs of machines, the chambers containing the end windings are normally under pressure which resists the flow of any extinguishing gases into such chamber, unless fed in at the ventilating fan or fans. In one case which I have in mind, where a fire occurred in the end winding, an attempt was made to introduce gas into the end housing, but without success, on account of the high internal pressure. The only effective extinguisher, in this particular case, was the use of a fire hose with the city water pressure behind it. This put out the fire, but the damage was almost as great as if no extinguisher had been used.

It should be remembered that the fumes from the fire are, in themselves, a good extinguisher, and therefore, if these fumes can be kept inside the machine, combustion will either be stopped or very much retarded. Therefore, one remedy which has been proposed at

times is not only to close the inlet in case of fire, but also to close the outlets. This may be difficult to accomplish practically, but should be fairly effective.

In the usual turbo-generator, the ventilating fans form part of the unit itself, and the ventilating tendency or action is present until the machine is shut down. However, in some recent machines of very large capacity, separate blowers are provided, and where such is the case, some automatic means might be provided for shutting down the blower when a fire is started in the machine. In such an arrangement, the effect of leaks would be very much less pronounced than in machines with self-contained blowers. However, a note of warning should be sounded in connection with all automatic devices which affect or control the ventilation of large turbo-generators. I refer to the fact that such devices, if they act improperly, may cause the very trouble which they are intended to remedy. Large turbo-generator units are so dependent upon artificial ventilation that they may overheat very quickly, and thus cause a fire, in case the ventilation is accidentally shut off, even for a short time. Therefore, if the fusible links mentioned by Mr. Lawler should act, through some defect, to close the inlet passages, or cut off the normal ventilation, and this were not discovered quickly, overheating might result.

Obviously, with such automatic devices, a signaling or indicating system should be used, which at once will call attention to the fact, when the air supply is cut off. So important is the supply of air to such machines under normal operation, that it is recommended, where separate ventilating fans are used, that these be driven by motors operated directly from the machines themselves, so that whenever the voltage is on a machine, its blower will automatically be kept in operation. One interesting feature in such an arrangement is that, in case of a short circuit, either inside or outside the machine, which tends to kill the voltage, it also tends to shut down the blower. In this way the power is automatically cut off the ventilating fan.

In conclusion I may say that, at present, the problem of protecting such a machine is one of completely closing the inlets or the outlets, or preferably both. If power station designers would give as careful consideration to these points as is given to many other points in station design, the risk of fire damage in turbo-generators could be greatly reduced.

I. E. MOULTROP (written). Mr. Lawler's paper points out some of the inherent defects of modern turbo-generators, but I question the value of the remedies he proposes. Modern turbo-generators run at a high rate of speed, the windings are as compact as possible, the windage of the machine is high and the insulation is quite inflammable; consequently there is a remarkably short interval between the time when the disturbance first starts and when the major portion of the exposed surface of the insulation is on fire. As soon as this takes place, the damage from an electrical standpoint has been done. I believe the practical solution of this problem is to bring all possible influence upon the manufacturers to devise and install non-combustible insulation, and I understand that there is some prospect of such insulation being brought out in the near future.

PAUL M. LINCOLN. Mr. Lawler says that the short circuit current of generators rises, in some cases, to as high as forty times the full load current. Although some turbo-generators have a short circuited current as high as that, it is not true of the modern turbo-generator. The designs put forth today will have a short circuit current of a value of somewhere between ten times and twenty times full load. This result is obtained by designing the generator so that the inherent reactance of the generator windings is higher than it is in slow-speed generators.

There have also been marked improvements in the method of bracing the generator winding so as to prevent motion under the tremendous mechanical strains that take place therein. I believe that I am perfectly safe in making the statement that so far as the protection against a mechanical stress due to short circuit is concerned, the modern generator does not require the addition of choke coils in order to make it safe, as suggested by Mr. Lawler.

I should like to ask Mr. Lawler whether it is the practice of the fire underwriters to insure generator windings against fire hazard originating within themselves? It is possible and in fact is standard practice, to install generators so that the windings can be completely consumed without any great fear of the fire communicating to the other parts of the building. So that a fire, if it does occur, may simply destroy the windings of the turbo-generator itself, without any great risk of its communicating anywhere else.

FORREST E. CARDULLO. It has been suggested that carbon dioxide be introduced into the air ducts of a turbo-generator for the purpose

of smothering a fire, but it seems to me that it would be more practicable to introduce steam for this purpose. To do so, it might be convenient to bleed the turbine at a point where the pressure is 15 to 30-lb. gage, and to introduce the steam into the air duct through some form of diffuser. An equally satisfactory method would be to pass steam from a saturated steam line through a reducing valve. The control of the steam might be automatic, so that the steam would be introduced by the melting of a fusible link, in the same manner as is proposed for the closing of the damper. I would call attention to the fact that the fusible links ought to be placed in the exit duct as close as possible to the generator; on this account, it would probably be better to have the damper placed in the exit duct than in the inlet duct.

H G. Reist (written). Having been intimately connected for many years with the design and construction of turbo-alternators, I have found this paper very interesting, and, in general, I agree with the views set forth in it.

It is true that some protection can be gained by the use of fire-resisting materials on the end of windings, but we must not expect too much from this because most varnishes are more or less combustible, and the fire-resisting materials, such as asbestos or mica, are too much divided to offer a complete barrier against the flames passing through them. At one time my attention was called to the fact that a cement floor when soaked with oil burns as well as a wooden construction, and the same holds true of the insulating material on turbo-windings.

It has been pointed out that it is difficult to introduce an inert gas into turbo-generators. This difficulty could, however, be overcome by introducing the gas in front of the fan, instead of beyond it, so that it would be drawn in by suction.

Another possible means of protecting these machines is to arrange a damper in such a way that it will not only close the inlet, but short circuit the outlet to the inlet, so that the additional air taken in will be air which has already passed through the generator, and, therefore, has its oxygen consumed. In European practice, where cotton filters have been used universally, there is danger of fire, and dampers have lately been introduced. How effective they are I cannot say. I have constructed dampers in some cases for turbo-generators, as long as six or seven years ago, but have not felt their utility sufficient to warrant the increased expense of using them generally.

So great have been the improvements in turbo-generators that fires are rare. Furthermore, the enclosed construction used on this type of machines makes it practically impossible for the fire to spread to the building.

Selby Haar. With regard to non-combustible coverings on the windings, it is perfectly feasible to get a non-combustible insulation, but the trouble is that non-combustible materials are usually pretty good heat insulators, and keeping the heat in his machine is just exactly what the designer is trying to avoid.

Albert Blauvelt. Judging from the preceding discussion there is a difference of opinion and a lack of real knowledge as to how quickly the copper gets hot and how much copper is really affected as compared with the whole of the copper. Mr. Lawler's fourth suggestion for the introduction of a dead gas indicates that the arcing does not usually immediately heat so much copper as necessarily to fry the entire machine before the copper could cool. On this assumption I would make two suggestions: (*a*) that a more prompt action at a lower heat can be secured by an ether pulse glass, or several of them, than by a fusible link or links; (*b*) that chimney flue gas is a good dead gas always available in large quantity, for example, as used by the Union Carbide Company. The particular details of solenoids, and relays, of dampers, cooling tank, pressure blower, exciting circuit breaker, etc., are a matter of expense and should or should not be gone into depending on whether the annual direct and indirect losses through these particular fires do or do not exceed 20 per cent on the cost of the protective equipment necessary to cover all the machines in service.

Roger D. De Wolf. In view of a recent experience in the burning out of a machine, I should like to hear from Mr. Lawler as to just how much he thinks the different methods suggested of cutting off the air supply and introducing carbon dioxide would reduce the damage to the machine. As pointed out by Mr. Moultrop, the time element is very short and there are a lot of things which might be done if there was time to do them, but in actual practice I believe the operating force would never get them done. In our case the short circuit resulted in burning off the main leads to the machine, and even if the air supply had been shut off, or the interior of the machine filled with carbon dioxide, I think the damage would not have been very greatly decreased. The amount of heat generated in the machine in

a very short period of time is so great that the carbonization of the insulating material would be carried to quite an advanced stage, and although the actual combustion might not take place on the exterior surface, the insulation on the windings would be practically destroyed, irrespective of whether there is an atmosphere around those windings which would support combustion or not.

THE AUTHOR. While it may be, as Mr. Lamme states, that there is no experience as yet showing that old machines are more liable to fire damage than new ones, it must be remembered that but very few turbo-generators can yet be considered old. This type of machine has only just passed through the development stage and therefore many new ones have given trouble, sometimes resulting in fire. In the future less trouble may be expected with new machines, but as they get older, the ordinary indications of age, such as short circuits, etc., may be looked for. It is reasonable to suppose that most machines will not be placed until one or more short circuits have occurred, and owing to the condition pointed out by Mr. Lamme and by myself in the paper, fires will probably occur in many instances.

It is undoubtedly important that means be taken to keep air leakage down to an inappreciable amount, so that the atmosphere in the machine will not support combustion after the carbon dioxide gas is introduced. Some leakage would of course be expected but to offset it, the carbon dioxide gas should be admitted in abundant quantities. It is not necessary that all gas within the generator be carbon dioxide that a fire be extinguished, for I understand that a mixture consisting of 25 per cent carbon dioxide and the remainder air, will form an atmosphere in which fire cannot exist. The exclusion of air and admission of the inert gas are details which can undoubtedly be satisfactorily worked out.

Regarding Mr. Lincoln's question as to the practice of fire insurance companies covering damage by fire to electrical apparatus when fire starts from electrical causes within the apparatus, I cannot answer for all companies, but it is my belief that policies very seldom cover such loss. The fact that policies do not cover such loss makes it even more important from the standpoint of the owners of turbo-generators that fires be prevented in their machines.

Professor Cardullo suggests the use of steam for extinguishing generator fires. Steam may have the advantage that large quantities will be quickly available, but it has the disadvantage that it will

thoroughly wet down the windings and in the case of old machines it might be difficult to avoid rewinding them, if they are wet down.

I have recently heard of some experiences which would indicate that steam was not as effective in extinguishing fire in enclosed places as has generally been thought. Carbon dioxide gas would extinguish fire more efficiently than steam, and being dry would not of itself cause injury to the machines.

In reference to Mr. DeWolf's question, the methods suggested in the paper are only to prevent occurrence of, or damage by, fire to the machine. They would not have any effect on the possible roasting of the insulation by continued heavy currents flowing in the winding. Short circuits have occurred in turbo-generators many times and probably most of them have not resulted in fire damage. It is probably a fact that where fire has not resulted, repairs to but a portion of the coils have usually been necessary and this would indicate that only in a small proportion of cases has it been necessary to rewind machines, due to a general roasting of insulation from the passage of current itself.

If the operators open the switches within a reasonably short time, as has appeared to be true in most cases in the past, roasting by current would not occur and it would appear well worth while to save the insulation from fire damage.

No. 1421

# EXTINGUISHING OF FIRES IN OILS AND VOLATILE LIQUIDS

Edw. A. Barrier,[1] Boston, Mass.

Non-Member

The extinguishing of fires in oils and in most of the volatile liquids has always been a difficult problem and where fires of this kind occur the results are frequently very disastrous. Our most common extinguishing agent, water, works rather unsatisfactorily upon the majority of such fires, but it is still the only one available where heroic measures are required. Comparatively recently, however, there have been two or three other materials introduced for use as extinguishers which have shown some promise for dealing with these fires, and it is the purpose of this paper to discuss these materials and the conditions under which they prove the most efficient.

2 Not all fires in volatile liquids are difficult to handle with water. When the liquid is miscible with water this extinguishing agent can be successfully used. Examples of this kind are denatured alcohol, wood alcohol, grain alcohol, acetone, etc. Where the liquid is not miscible with water little or no effect is produced except to wash the burning liquid out of the building where it may be completely consumed or, if the quantity of oil is small, possibly to extinguish the fire by the brute cooling effect of a large quantity of water sprayed upon the fire. Soda and acid extinguishers are somewhat more effective than pure water, but even they fail under most conditions. The various grenades containing salt solutions which were formerly extensively exploited are of course practically worthless.

3 The only principles that can be made use of in extinguishing fires in volatile oils are, (*a*) to form a blanket either of gas or of solid material over the burning liquid which will exclude the oxygen of the air, or, (*b*) to dilute the burning liquid with a non-inflammable extinguishing agent which is miscible with it.

[1] Inspection Department, Associated Factory Mutual Fire Insurance Companies, 31 Milk Street.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

SAWDUST AND BICARBONATE OF SODA

4 To the blanketing type of extinguishers belongs sawdust. Paradoxical as it may seem, ordinary sawdust is an excellent extinguishing agent for certain volatile liquids, especially those of a viscous nature. A considerable number of experiments were conducted in the fall of 1912 by the inspection department of the Associated Factory Mutual Fire Insurance Companies, in the extinguishing of fires in lacquer and gasolene in tanks with sawdust, and the results were surprisingly satisfactory.

5 The liquids were placed in three tanks 30 in. long, 12 in. wide and 16 in. deep; 48 in. long, 14 in. wide and 16 in. deep; and 60 in. long, 30 in. wide and 16 in. deep. The sawdust was applied with a long-handled, light but substantially built snow shovel having a blade of considerable area. In every case the fires were extinguished readily, especially in the two smaller tanks which were about as large as any ordinarily employed for lacquer in manufacturing establishments.

6 The efficiency of the sawdust is undoubtedly due to its blanketing action in floating for a time upon the surface of the liquid and excluding the oxygen of the air. Its efficiency is greater on viscous liquids than on thin liquids, since it floats more readily on the former than on the latter. The sawdust itself is not easily ignited and when it does become ignited it burns without flame. The burning embers have not a sufficiently high temperature to reignite the liquid.

7 The character of the sawdust, whether from soft wood or hard wood, appears to be of little or no importance, and the amount of moisture contained in it is apparently not a factor, so that the drying out of sawdust when kept in manufacturing establishments for a time would not affect the efficiency.

8 It was found that the admixture of sodium bicarbonate greatly increased the efficiency of the sawdust as shown both by the shortened time and the decreased amount of material necessary to extinguish the fires. A further advantage of the addition of bicarbonate of soda is that it decreases the possible danger resulting from the presence of sawdust in manufacturing plants since it would be difficult, if not impossible, to ignite the mixture by a carelessly thrown match or any other ready source of ignition.

9 Although the efficiency of the sawdust is greatest on viscous liquids such as lacquers, heavy oils, japan, waxes, etc., in the tests referred to, fires were extinguished in gasolene contained in the

smallest tank and also when spread upon the ground. In larger tanks the sawdust or bicarbonate mixture does not work so well since the sawdust sinks before the whole surface can be covered, whereupon the exposed liquid reignites.

### CARBON TETRACHLORIDE

10 In recent years carbon tetrachloride has received considerable attention as a fire-extinguishing agent. This is due largely to the activity of certain manufacturers of fire extinguishers which use liquids, the basis of which is carbon tetrachloride.

11 This substance is a water white liquid and possesses when pure a rather agreeable odor somewhat similar to chloroform. A considerable proportion of the commercial article upon the market, however, contains sulphur impurities which impart a disagreeable odor to the liquid. The substance is quite heavy, its specific gravity being 1.632 at 32 deg. fahr. It is non-inflammable, non-explosive, and is readily miscible with oils, waxes, japan, etc. When mixed with inflammable liquids it renders them non-inflammable provided a sufficient quantity is added. Its vapor is heavy, the specific gravity being about five and one-half times that of air, consequently it settles very rapidly. As an extinguishing agent it operates by both the principles mentioned in Par. 3, namely, it dilutes the inflammable liquid rendering it non-inflammable, or at least less inflammable, and it forms a blanket of gas or vapor over the burning liquid which excludes the oxygen of the air.

12 Although this paper is confined to a discussion of extinguishing fires in oils and volatile liquids, it may not be out of place to mention that the claims made by certain manufacturers producing these extinguishers which use liquids, the basis of which is carbon tetrachloride, are grossly exaggerated. These preparations, none of which is more efficient than carbon tetrachloride, are not the equivalent of the ordinary water extinguishers for general use on such materials as cotton, wood, paper, oily waste, etc.

13 On volatile liquids, oils, etc., carbon tetrachloride has, however, shown very satisfactory results under some conditions, but the readiness with which a fire can be extinguished with it depends to a considerable extent upon the skill of the operator and the nature of the fire. In tank fires the length of time that the liquid has been burning is an important factor, and in such cases where the sides of the tank become heated the only way in which the fire can be ex-

tinguished is to squirt the liquid forcibly at the sides. If the carbon tetrachloride is squirted directly into the liquid it is much more difficult, if not impossible, to extinguish the fire.

14 The height of the liquid in the tank is also a very important factor. Where the liquid is low the sides form a pocket which retains the vapor and aids considerably in smothering the blaze. When the tank is nearly full, however, this condition does not exist, and it is then very difficult, if not impossible, to extinguish a fire in a highly volatile liquid, such as gasolene; only the most skilled operators are successful in these cases. The size of the tank or the extent of the fire if upon the floor is, as would be expected, of considerable importance. In tanks larger than about 28 in. by 12 in. more than one extinguisher and operator working at a time are necessary to extinguish a fire in such materials as gasolene. In one test where a tank 60 in. by 30 in. was used no less than seven operators were necessary, and even then it was only with the greatest difficulty that the fire was put out.

15 All of the above remarks apply to carbon tetrachloride in the ordinary one-quart extinguisher as generally sold. It is probable that a large extinguisher which could throw a large stream would prove more efficient, but on account of the great weight of carbon tetrachloride such an extinguisher would have to be specially designed to make it readily portable by mounting on a truck or some similar means. Expelling the liquid by means of a hand-pumping arrangement would probably be unsatisfactory, and it would therefore be necessary to force it out in some other way.

16 A few systems have recently been installed in which an elevated tank containing carbon tetrachloride was connected with automatic sprinklers or perforated pipes located in hazardous rooms where volatile and inflammable liquids are in use. So far as is known none of these systems have as yet been called upon to extinguish a fire, but there appears to be no reason why such a system should not provide excellent protection in special cases. In such systems it would be necessary to consider the safety of the workmen and furnish ready means of escape, since carbon tetrachloride is an anesthetic and where thoroughly sprayed through the air as from an automatic sprinkler it would probably produce rapid results.

17 The nature and effect of the fumes given off when carbon tetrachloride is thrown upon a fire is a subject which has received a great deal of discussion. When the liquid comes in contact with a

fire the vapor is partly decomposed resulting in the evolution of a considerable quantity of black smoke which is undoubtedly finely divided carbon. Pungent gases are also produced which appear to be mostly hydrochloric acid with possibly a small amount of chlorine. Since carbon tetrachloride contains no hydrogen from which hydrochloric acid could be formed this substance must be produced by the action of chlorine on the gases arising from the burning material or upon the moisture of the air.

18 The fumes of carbon tetrachloride although of a very pungent nature do not produce any permanent injury under ordinary conditions where the operator can make his escape after he has inhaled all that he can stand, but they are a distinct handicap in fighting a fire and are one of the objectionable features to carbon tetrachloride as a general fire extinguishing agent. In large rooms or where a small quantity of carbon tetrachloride is sufficient to extinguish a fire the gases are of course less objectionable.

### FROTHY MIXTURES

19 Another method of extinguishing fires in oils and volatile liquids which has recently been proposed and experimented with is that of using frothy mixtures. The idea seems like a very promising one and the tests which have been thus far reported indicate very satisfactory results. The idea was originated and has been developed in Germany. So far as is known no experiments have been conducted in this country.

20 The process consists essentially in causing two liquids to mix in a tank where foam is produced. The tank is made airtight and sufficiently strong to permit of the foam being forced out by carbon dioxide under pressure, and the foam is conveyed to the fire by means of a line of hose. The exact nature of the liquids has not been disclosed, but one of them probably consists of a sodium carbonate solution containing froth-forming ingredients such as glue or casein and the other an alum solution. The two on coming together generate carbon dioxide which produces froth. This froth is reported to be quite stiff and to shrink in volume but a comparatively small amount even after a period of half an hour.

21 A number of tests were conducted in the winter of 1912 in Germany; some of them on a considerable scale. In one case as much as 5 tons of crude naphtha in a tank was involved, and in another an area of 1300 sq. ft. of burning tar was used. In all cases the results

were reported satisfactory, the fires being extinguished in a short time.

22 The frothy mixture undoubtedly owes its efficiency to its blanketing action in settling upon the surface of the burning liquid, thus excluding the oxygen of the air, and to the fact that the bubbles of liquid contain carbon dioxide which upon bursting produce an atmosphere in which combustion cannot take place.

23 According to the latest reports the matter is still in an experimental stage, various details regarding the form of apparatus, most efficient pressure, and design of nozzles being under consideration; but from what has already been done it would appear that the idea is a very promising one, and that this method of extinguishing fires in oils and volatile liquids will prove to be by far the most efficient of any that has as yet been suggested.

## DISCUSSION

A. E. Cluett. As I understand it, the tank was an open one, of rectangular section, which does not represent the conditions in which these liquids are ordinarily found. They are generally in a closed tank, with a bung-hole of some description, and I would like to ask whether any experiments have been made on such receptacles, and also whether experiments have been conducted where the fluid has been spread out as it would be if it were spilled over a floor.

J. Stewart Thomson (written). The author of the paper occupies a position in connection with an organization insuring sprinklered risks, which makes no recognition whatever of devices for the extinguishing of fires other than by sprinklers. His remarks are the result of some tests made by him in connection with devices other than sprinklers.

It is generally known that fires in volatile liquids which are not miscible with water cannot be extinguished by the application of water. The reason is perfectly apparent; and that the way to extinguish a fire in volatile liquids of this type is to blanket the flame or to dilute the liquid with a non-inflammable extinguishing agent, thus cooling the liquid to the point where it will not burn.

The use of sawdust in connection with fires of this character is not new. Almost every one has at one time or another used a blanket, towel or something similar for the purpose of smothering a fire. We all know that sawdust will float for a time at least on the surface of

the liquid, and that if the sawdust is put on in quantity it acts as a blanket. As a practical fire extinguishing agent, however, the use of sawdust is not practical. In a large tank of burning liquid of the type described, before the surface can be covered with the blanket, much of the sawdust will sink below the surface whereupon the exposed liquid reignites.

As to the use of carbon tetrachloride as a means of extinguishing fires in oils and volatile liquids, there are some 40 different devices on the market employing this substance. Carbon tetrachloride has been found to be remarkably efficient; a quart of this liquid contains approximately 149 cu. ft. of fire extinguishing gas. As Mr. Barrier states, carbon tetrachloride contains sulphur impurities. Its use in a metal container results after a period of time in the corrosion of the metal. Extinguishers generally employing this chemical have not been found to have long life and corrosion of the metal results in the clogging of the working parts of the extinguisher. These extinguishers have the additional drawback of operating by air pressure; it is a well-known fact that air cannot be contained over any very long period in a metal container with valves, outlets, petcocks, etc.

The manufacturers of the Pyrene extinguisher discovered the efficiency of carbon tetrachloride as an extinguishing agent many years ago, but also that this article is corrosive in its action upon metals. It was necessary to alter the nature of carbon tetrachloride in respect to this before it could be used successfully in a fire extinguisher. They also discovered that it was possible to increase the efficiency of the liquid.

While the claim is not made by the author, the inference would be from his paper that carbon tetrachloride is virtually the same as the extinguishing agent employed by the Pyrene Manufacturing Company, in his reference to "the activity of certain manufacturers of fire extinguishers which use liquids, the basis of which is carbon tetrachloride." He is probably not informed that one quart of Pyrene will generate approximately 33 $^1/_3$ per cent more fire extinguishing gases than carbon tetrachloride; therefore it will cover more fire. The one-quart pump-type Pyrene extinguisher has been approved by the Chicago Laboratories, Inc., and is in universal use throughout the country; is uniformly recognized in every state for fires of the type described by him, and that its record of efficiency in respect to these fires is remarkable.

In his statement in Par. 12, Mr. Barrier ignores the amount of

extinguishing gas referred to above. Extinguishers of this type are more efficient in cotton fires and fires in oily waste, etc., than those employing solutions of water.

In the test where a 60-in. by 30-in. tank was used, where it required a number of operators to extinguish a fire of burning gasolene, he should certainly have had no difficulty in extinguishing a fire in a tank of these dimensions if the extinguisher had been properly handled.

Mr. Barrier concludes that the use of soapsuds, described as "frothy" mixtures in extinguishing fires in oils and volatile liquids, will prove by far the most efficient of any that has yet been suggested, but statistics will not bear out the conclusion. This method has been employed to some extent on the Continent and in England for some time, and experiments have also been conducted in this country which are the subject of extensive reports. The progress of extinguishers working on this principle, however, has been inconsiderable as compared with the progress of other extinguishers, and notably, as compared with the progress made by the Pyrene extinguishers.

Henry W. Appleton. I would ask Mr. Barrier what he considers the proper proportion of soda carbonate in the sawdust, and what effect is produced on the sawdust to make it non-inflammable; that is, not only the percentage to be used to make it efficient in putting out the fire, but also the percentage to be used in rendering the sawdust itself non-inflammable.

Lewis H. Kunhardt. Mr. Thomson's statement, that the author occupies a position in connection with an organization insuring sprinklered risks, which makes no recognition whatever of devices for the extinguishing of fires other than by sprinklers, needs modification. It is absolutely at variance with the fact. Not only do the Mutual Companies give credit for many, and practically all devices which are used in extinguishing fires, but all the various sprinkler risk associations, underwriters and others throughout the country, are continuously studying all these other devices. Sprinklers are only a part of the fire protection of a piece of property. If other devices fail, the sprinklers come in and assist, and if the sprinklers should fail, there are the fire pumps, hose, etc. All the various fire protection engineers in the country are studying these problems and details thoroughly, and they do not give recognition *only* to the automatic sprinkler, although they recognize the value of it, and know it to be one of the best fire fighting agencies of today.

Albert Blauvelt. As Mr. Kunhardt comments, this paper by Mr. Barrier covers a subject in itself and one which is not tied to automatic sprinkler practice. The use of hand apparatus is universal in all branches of fire protection practice in preference to waiting for sprinklers to fuse, or to waiting for hose, and the greatest success of the professional fire department so far as the majority, or some 70 to 80 per cent of the fires numerically are concerned, is accomplished with hand appliances, or usually the carbonic-acid gas extinguishers carried on the man's back as he jumps off the hose cart.

The general scope of hand apparatus is fairly broad, depending on the combustible to be dealt with and whether ceiling as well as floor fires are likely to need extinguishment. Under its title, the teaching of Mr. Barrier's paper would seem to be that this subject of handling oil or volatile or chemical fires is not a mystery. Doubtless, as suggested by a preceding speaker, the paper can be bettered by giving more detail as to the best proportions for the several useful mixtures, but the main thing is that those who will can make up effective extinguishing compounds for volatile liquid or oil fires, not perhaps of the exact alchemic virtue of the trade compounds but probably obtain twenty times the quantity at like cost.

Gorham Dana.[1] This paper seems to cover the field remarkably well, but there is one point I would like to bring up in connection with carbon tetrachloride, and that is the effect which the gas might have on metals in the vicinity. At a fire some time ago in a garage, an attempt was made to extinguish a pail of burning gasolene with these extinguishers. They not only failed to extinguish the fire, but the gases given off were of such a corrosive nature, probably hydrochloric acid, that the damage done to the metal work on the automobiles and in the garage, was more than the direct loss from the fire itself.

F. E. Cardullo. The claim has been made in this discussion that one quart of carbon tetrachloride at atmospheric pressure liberates a volume of 149 cu. ft. of inert gas. A hasty calculation based upon my memory of the density of air gives for the volume of carbon tetrachloride vapor liberated from one quart of liquid a value of approximately 8 cu. ft. From the data given by Peabody in his steam tables, the volume of saturated vapor liberated at atmospheric pressure from

[1] Manager, The Underwriters' Bureau of New England, 141 Milk Street, Boston, Mass.

one quart of carbon tetrachloride is 9.97 cu. ft. From data based upon the theoretical vapor density of carbon tetrachloride the volume of the vapor generated from a quart of the liquid at a temperature of 135 deg. fahr., and a pressure of one atmosphere, is 9.8 cu. ft.

It may be added that if the gas given off occupied a volume of 149 cu. ft., it would be so light as to rise in the air and would have no smothering effect upon the fire. I would suggest that, in discussion, our members ought to be exceedingly careful of their facts, and not to quote any figures, or to make any statements, of the truth of which they are not absolutely certain.

The Author. Replying to the question of Mr. Cluett, all the experiments with these materials were tried in open containers, since that is the way lacquer is ordinarily used in most metal-working plants. The metal parts are generally dipped into the lacquer. No tests were tried on closed tanks, and it would be difficult to use sawdust and bicarbonate of soda in such a tank. It might be possible to introduce carbon tetrachloride under such conditions. One of the tests did include a fire in lacquer spread over a wooden platform which served as a floor, and the sawdust appeared to work very satisfactorily in this case.

With reference to the comments of Mr. Thomson, I would say that I recognize that sawdust as an extinguishing agent is not new, but it is not well known, and this fact is shown by the experience we have had where this material has been recommended to manufacturing establishments throughout the country. At first their attitude was one of great surprise but after they had conducted a few tests of their own, they were very well pleased with the results. This shows that even though the idea may not have been new, it was at least not at all well known and the feature of mixing bicarbonate of soda with the sawdust is a new and valuable one.

It was not my intention to mention any particular commercial device, but in view of the fact that the Pyrene extinguisher has been brought into the discussion, I will say that with reference to the matter of the corrosive action of carbon tetrachloride, tests were conducted in a Pyrene extinguisher with an ordinary garden variety of carbon tetrachloride, the usual commercial article, in which the liquid remained in the extinguisher for over a year, but as far as could be seen after using a can opener there was absolutely no indication of corrosion. In certain cases where moisture or hydrochloric acid are

present, carbon tetrachloride does cause some corrosion, but the indications are that if a reasonably pure grade of the material is obtained, there is no serious trouble from this source.

As to the comparative efficiency of Pyrene and carbon tetrachloride, a large number of tests were made to compare the two liquids when both were used under the same conditions. As far as a number of unprejudiced observers could determine, there was no difference in the efficiency of the two. It is true that Pyrene contains a few other substances besides carbon tetrachloride, among which is a small amount of gum, which by the way is combustible, and a small amount of nitro benzol which is also combustible; both of these are detrimental rather than advantageous as far as extinguishing a fire is concerned.

With reference to the proportions of bicarbonate of soda and sawdust, the ingredients are mixed in the proportion of about 8 lb. of bicarbonate of soda to a bushel of sawdust. This proportion does not render the sawdust non-combustible but it does retard its ignition. It works in the following way: If a spark or any other source of ignition comes in contact with the mixture in the vicinity of where the bicarbonate of soda is located, the latter becomes heated and begins to give off carbon dioxide. This tends to make the atmosphere in that vicinity a non-supporter of combustion and serves to extinguish or retard the fire.

In reply to the question by Professor Woolson as to whether there is any tendency toward caking after the sawdust and the bicarbonate of soda have been standing for any length of time, I may say that this feature has been pretty thoroughly tested under the worst possible conditions and after a period of six or eight months hardly any indication of caking could be detected.

# No. 1422

# A SYSTEM FOR THE CONTROL OF AUTOMATIC SPRINKLER VALVES

By Fred J. Miller, New York

Member of the Society

The fires in the Triangle Shirtwaist Factory and in the Binghamton Clothing Company's building, have served strongly to emphasize the fact that public sentiment is more and more placing full responsibility upon the owners of industrial establishments for the thorough protection of employees from death by fire. This responsibility is being reflected in legislation requiring the provision of suitable means of egress from buildings, fire drills, fire walls, automatic sprinklers, enforcement of rules against smoking, etc.

2 The outside fire escape, constructed in the ordinary manner, was proved in both the fires mentioned above and in numerous other fires as well to be almost, if not absolutely worthless. It is extremely difficult for a large number of employees, especially where many of them are women, to descend the ordinary fire escape, even when there is no panic and no reason for special haste. At the time of a fire it is practically useless.

3 Fire drills are probably justifiable and have in many cases saved lives. If properly maintained they are expensive, however, and in both the fires referred to above they proved worthless.

4 Some of the other things mentioned, particularly suitable fire walls, are very good, but experience seems to have proved abundantly that the best of them all is the automatic sprinkler. In theory, at least, it is supposed to be always ready for instant use, and it puts water where and when it is needed at the very beginning of a fire, before it has had time to assume threatening proportions, and usually before there is enough smoke to alarm the occupants of the building seriously or start a panic.

5 It has other important uses, such as the formation of an effective water curtain capable of arresting the progress of a fire which has

---

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

assumed the character of a conflagration, and Mr. Albert Blauvelt and Prof. Ira H. Woolson in their papers read at the Baltimore meeting of the Society[1] present much impressive evidence of the value of sprinklers when maintained in such working order as to be available when needed.

6 The National Fire Protection Association, through its committee on safety to life, investigated the Binghamton Clothing Company's fire and in their report, commenting upon the various conditions in the building and particularly upon the entire absence of practically every efficient safeguard, say:

Long experience and continually repeated demonstration prove that the automatic sprinkler, where properly installed and in operating condition, is always ready, operates quickly, and either extinguishes fire or holds it in check, and is the most reliable means of safeguarding lives in the majority of existing manufacturing buildings. Enclosing stairs, building fire towers, properly constructing fire escapes, etc., are all means intended to permit of escape after the fire gets under way. The automatic sprinkler almost invariably prevents the fire from assuming serious proportions.

7 This statement by an association which has made a very thorough study of the whole problem of the lessening of fire loss and the protection of lives from destruction by fire, is borne out by the experience of the Manufacturers' Mutual Fire Insurance Companies. Every risk covered by these companies must be protected by automatic sprinkler heads. It is impressive to note that the 19 companies, comprising what is known as the senior group, carry policies on industrial establishments aggregating over $2,500,000,000; employed in these factories there are, as nearly as can be ascertained, about 2,000,000 people, and so far as known not a single life has been lost in any of the buildings thus insured where the sprinkler heads were in commission at the time of the fire. A few lives have been lost in these factories, but in every such case the fire gained headway because, for one reason or another, the automatic sprinklers were out of commission at the time.

8 The general conditions respecting factory insurance are apt to surprise those who investigate them for the first time. Among the features to be discovered is that speaking broadly the stock insurance companies do not require sprinkler heads, nor more than a fractional part of the other safeguards that are required by the Factory Mutual Companies. In recent years their leading men have strongly recommended sprinklers and they give great concessions in rates upon sprinklered risks, but the local agent of an insurance company, work-

[1] Trans. Am. Soc. M. E., vol. 35, pp. 171, 231.

ing for a percentage upon the premium, virtually recommends that his own income be greatly reduced when he urges sprinklers. Their business is organized mainly with a view to simply carrying at a profitable premium rate the fire risk as it is, and save through such agencies as their Factory Insurance Association and certain committees of limited scope, they do little to lessen the fire risk and thereby the danger of loss of life.

9 In the case of a typical factory, for instance, where the local agent for a stock company receives $200 commission per year on the risk in an unprotected condition having combustible occupancy, he might receive only $10 if the same building were fully protected against fire by automatic sprinklers, etc. There is, therefore, clearly no incentive for this agent to inform the owner how his fire risk and the amount of premiums he pays for insurance may be reduced; in fact there is every commercial incentive in the opposite direction, until "mutual competition" threatens to take the business entirely out of his hands.

10 It is interesting to note that several construction companies are now offering to install automatic sprinklers without the payment of any cost whatever by the owner, the company making the installation offering to take as its pay the difference between the present insurance premiums and the lower premiums that will be paid after the installation is made, for a number of years to be determined upon. The meaning of this is simply that the owner of the building and its occupants are very much better protected against fire without any additional outlay, and that at the end of a few years the owner will have the fire protective system thus installed for his own, free of cost.

11 I do not intend to advise either for or against these companies, but cite the fact simply as conclusive evidence of the pecuniary as well as the other advantages of automatic sprinklers.

12 But sprinkler heads when once installed must then be kept ready for service when needed. Sprinkler heads are usually arranged in groups with a supply pipe for each group. In each of these main supply pipes is a valve by means of which water can be shut off from that section when necessary for repairs or by reason of the opening of one or more sprinkler heads, either in performing its normal service, or by accident. These valves cannot be locked open because it may be necessary in an emergency to close them as quickly as possible.

13 These valves introduce an element of danger and with this it is the main object of this paper to deal. The inspectors of the Mutual Companies frequently find the valves referred to closed, and the

sprinkler heads, therefore, of no more use than as though they had never been installed.

14 In an effort to lessen this evil, the Factory Mutual Companies occasionally send to their policy holders a circular letter on the subject of the danger of closed sprinkler valves. In such a report, covering the months of April, May and June 1912, 72 cases were reported where inspectors found such valves closed. In 63 of these cases the number of sprinkler heads controlled by the closed valves are given, and these foot up to 8274 heads. Assuming that the cases in which the number of heads is not given would average about the same, this means that for this report, covering three months, nearly 10,000 sprinkler heads were found out of use by reason of valves closed when they should have been open.

15 In many cases no reason was assigned for the valves being closed, which means, we may infer, that the proprietors did not know how the valves came to be closed. It is inconceivable, of course, that they would go to the expense of installing automatic sprinklers and other fire protective devices, and then allow them to be thrown out of use if they had the organization and the discipline necessary to keep such devices always available.

16 The Remington Typewriter Company owns and operates five factories on which it carries in the Mutual Companies insurance policies aggregating about \$6,000,000. These factories are protected by nearly 10,000 sprinkler heads, the water supply for which is controlled by 84 valves.

17 We have had from the start a system of inspection under which a watchman at frequent intervals examines each valve to see whether or not it is open. If found closed, the reason therefor is investigated, a report is made and an effort to prevent a repetition of the error.

18 For many years no inspector of the Mutual Companies had found one of our sprinkler valves closed, until about a year ago when one was found. The reason revealed by an investigation is probably typical of a good many such cases. A mechanic had been making some repairs in a boiler house, over the boilers, and under the roof where the sprinklers were. Before going up over the boilers he closed the valve controlling these sprinklers as a precaution against accidental flooding. When he had finished the work he called down to his assistant to open the valve. He took it for granted that the assistant had done so, but for some reason the valve was not opened and the next day the inspector arrived.

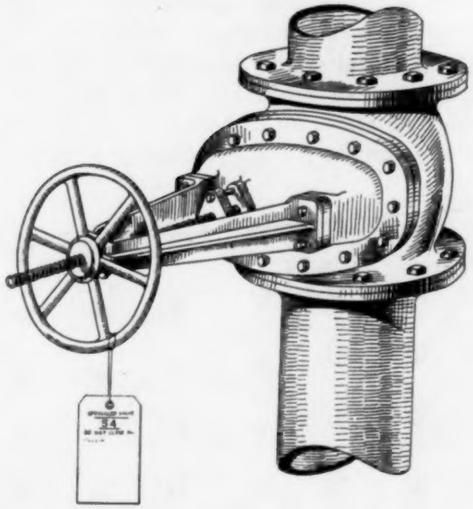


Fig. 1 Sprinkler Valve Open with Instruction Tag Attached

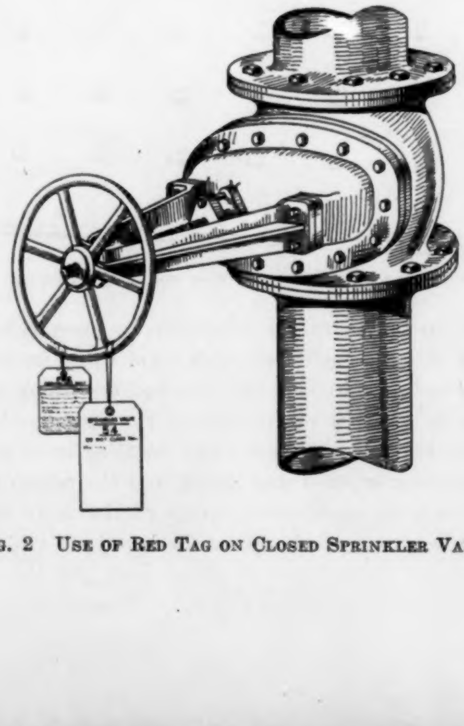


Fig. 2 Use of Red Tag on Closed Sprinkler Valve

19 To decrease the liability of valves being either closed without authority, or left closed when they should have been opened, we have devised and have in use in our factories the system to be described:

20 Our sprinkler valves are all numbered and upon each valve there is hung a tag as shown in Figs. 1 and 4. This tag is marked with the number of the valve to which it is attached, and, printed upon the tag is a prohibition of the closing of the valve without a

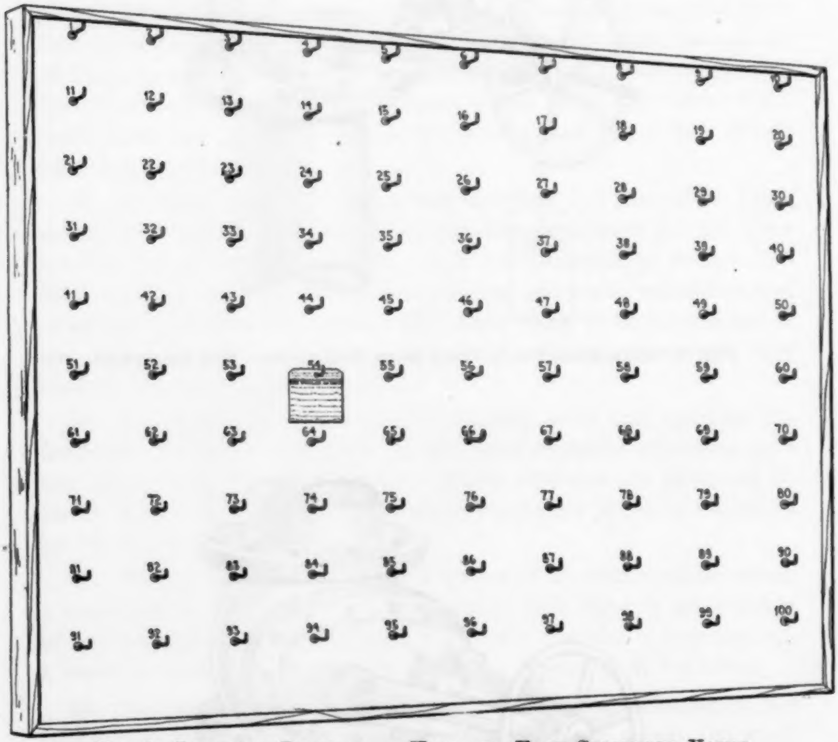
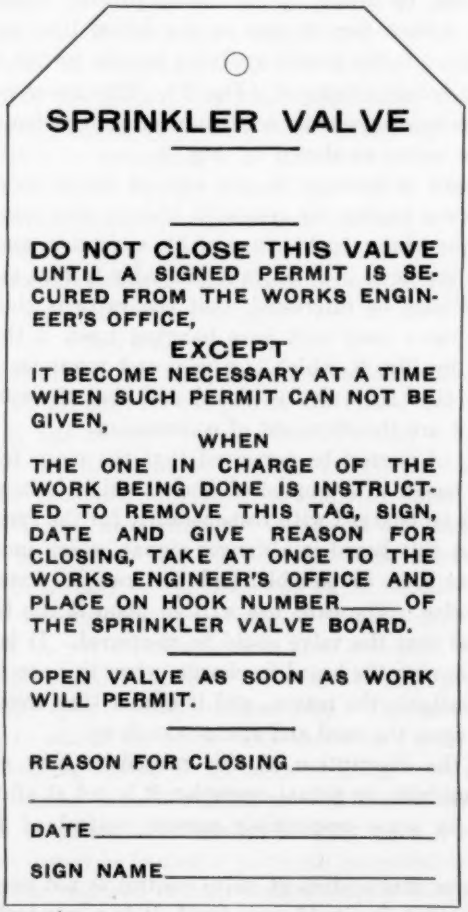


FIG. 3 INDICATOR BOARD WITH HOOK FOR EACH SPRINKLER VALVE

signed permit to do so, except when, in an emergency, or for any special reason it must be closed at a time when no permit can be obtained, in which case it is to be detached and hung upon a board, shown in Fig. 3. This is as directed by the card itself, and when it is done the blanks upon this card must be filled in to show by whom the valve was closed, when it was closed, and the reason for closing.

21 When such an emergency closing of the valve takes place, at a time when the person in charge of the board (Fig. 3) is away, the

watchman or other person having access to it goes to the board with the person who has closed the valve. Both these men then see that a red card complete but left blank is hung upon the board over the card removed from the closed valve. These are hung upon the hook


SPRINKLER VALVE

DO NOT CLOSE THIS VALVE
UNTIL A SIGNED PERMIT IS SECURED FROM THE WORKS ENGINEER'S OFFICE,

EXCEPT

IT BECOME NECESSARY AT A TIME WHEN SUCH PERMIT CAN NOT BE GIVEN,

WHEN

THE ONE IN CHARGE OF THE WORK BEING DONE IS INSTRUCTED TO REMOVE THIS TAG, SIGN, DATE AND GIVE REASON FOR CLOSING, TAKE AT ONCE TO THE WORKS ENGINEER'S OFFICE AND PLACE ON HOOK NUMBER_____ OF THE SPRINKLER VALVE BOARD.

OPEN VALVE AS SOON AS WORK WILL PERMIT.

REASON FOR CLOSING__________

____________________

DATE________________

SIGN NAME_____________

Fig. 4 Instruction Tag Attached to Every Sprinkler Valve

numbered to correspond with the number of the closed valve. These cards thus hung upon this hook are intended to attract the notice of the person having the board in charge, who will, after proper investigation, fill out the blanks on the red card, hang the lower portion of it on the numbered hook corresponding to the closed valve, and attach

the upper portion to the valve itself, as shown in Fig. 2, unless it shall have previously been re-opened.

22 In most cases it will be possible to obtain a written permit before closing a valve, and in such case the permit is given by the person in charge, by filling in the blanks printed upon the red tag (Fig. 5). It is then torn in two at the dotted line, and the upper portion of it given to the person applying for the permit, who attaches it to the valve before closing it (Fig. 2). The lower portion of the same red tag is hung upon the hook numbered to correspond with the number of the valve, as shown by Fig. 3.

23 The board is fastened to the wall of the room, or above the desk of the person having the system in charge, who may be the chief engineer, the superintendent, or one of his assistants, and the red tag hanging upon the hook is a constant reminder to him and a token to all others who may be interested, that the valve is closed. During this time the valve itself will have hanging upon it the red permit tag, as shown by Fig. 2, which is a sign and reminder so long as it remains there that the valve is closed and that the sprinkler heads controlled by it are therefore out of commission.

24 It will, of course, be perceived that the main feature of this system is the board with numbered hooks, which is in plain view of the factory official charged with responsibility for the sprinkler valves; a red tag upon this board is a danger signal, which means it should be taken off as soon as possible, and this can be done only by re-opening the valve. The card has written upon it the time at which it was expected that the valve could be re-opened. It is the business of the official having the board in charge, when this specified time has passed, to investigate the reason, and if a later time must be assigned to indicate it upon the card and again take it up.

25 While the description of this procedure may make it seem somewhat formidable, in actual operation it is not at all inconvenient, and it gives to some responsible person control of the sprinkler valves.

26 Of course this system of valve control is not presented under the impression that, by itself considered, it is a wonderful invention. So far as that is concerned it is no more than is being done every day for the accomplishment of various objects. It was devised to overcome a difficulty which, in view of all the conditions now existing, it seems rather important to eliminate in factories generally where sprinklers have been installed; it can save property from destruction by fire and may easily save the lives of those employed in factories; for

that reason it is here presented. It was not devised by myself alone, but, in accordance with the regular practice in our organization, the idea of it was presented to all our factories and it was then developed and perfected by the suggestions of the superintendents of those factories, Messrs. E. E. Barney, G. B. Brand, W. N. Brand, all of whom are members of the Society, and Mr. C. W. Burges.

27 Mr. John R. Freeman thought so well of it that it has been recommended by the Mutual Companies, including those of which he is president, and his associates have added some improvements to the red card, notably by arranging it so that the coupon which has been hung upon the board to signify a closed valve goes, when the valve is opened, to the superintendent, who is thus enabled to know how much of valve closing is taking place, and the reasons for it. This, I believe, is an excellent feature. The cards are now being supplied to all policy holders in the Mutual Companies and a reproduction of them is shown in Figs. 5 and 5a.

28 In some of our factories, there are certain valves which, with the approval of the mutual companies are closed during the winter months. As it was thought to be undesirable that a red card should hang upon the board, as shown by Fig. 3, for some months, because danger signals constantly displayed are apt to cease to convey the idea of danger, they have proposed in one of our factories to cover the red card hung upon the board under such circumstances by a green card. That may be the best way to do it, but I am inclined to think that a better way is to leave off the red card entirely from the board and to put a memorandum into a tickler file so that the matter of reopening the valve will be automatically presented to the person in charge upon a predetermined date. This memorandum may appropriately be one of the white cards shown in Fig. 4 properly filled out; but the red tag should in all cases be hung upon a closed valve.

29 Mr. Freeman does not think so well of the white card. We use it because by means of it, always hanging upon a valve, a person whose business it may be to close a valve can learn and become familiar with what else is required of him besides merely closing a valve and then perhaps forgetting it. In the interest of the life-saving feature, the Remington Typewriter Company will supply such cards free to all who may wish to use them. Some are printed upon cloth for use when valves are out of doors, others upon cardboard. With the cards thus supplied it will be necessary only for the factory to provide for itself, the board with ordinary screwed brass hooks, numbered as shown—to put the board up before the eyes of a responsible factory

(SEE OTHER SIDE)

# FIRE

## SPRINKLERS SHUT

**Attach this Tag to Valve**

Valve No......... { Every Sprinkler Valve should be numbered, preferably with white paint figures at least 2 inches tall. May be on pipe close to valve.

Shut on.................................191 ......M.
Day of Week Month Date Hour

By..................................................
Name of man who shut the valve

To be shut
Only until..............................191 ......M.
Day Month Date Hour

Reason for Closing..................................

....................................................

Closing Authorized by

Tag No........ .....................................
Signature of Master Mechanic or other official

**Don't remove this tag until valve is opened. If valve is not opened at above time, fill out and attach a second tag extending time; also a third tag if need be.**

Opened On...............................191 ......M.
Day Month Date Hour

By.......................... { After opening valve take this tag to Master Mechanic's Office to be kept until next Insurance Inspector's visit.
Signature of man who opened valve

....................................................

**This Coupon is to be torn off and hung on conspicuous peg board in Master Mechanic's Office until tag is returned signed by the man who opened valve. After return of tag, enter date of opening on this coupon, and send it to General Superintendent's Office.**

**FIRE SPRINKLERS ARE SHUT OFF**

In..................................................
Name of Building and room

At Valve No....... on...........................191 .
Day Month Date

By..................................................
Name of man who shut the valve

To be shut
Only until..............................191 ......M.
Day Month Date

Reason for Closing..................................

....................................................

Closing Authorized by

Tag No........ .....................................
Signature of Master Mechanic or other official

Opened by.......................... on..........191 .
Name of man who opened valve Date

(SEE OTHER SIDE)

Fig. 5 Form of Red Card used by the Mutual Companies

(SEE OTHER SIDE)

The danger of greatest loss from Fire in factories insured in "The Mutuals" to-day comes from Sprinkler Valves improperly closed or forgotten.

On almost every day the insurance inspectors find somewhere an important Sprinkler Valve shut. Often this occurs in factories which take pride in their order and discipline.

## RULES.

1. Sprinkler valves must not be closed without authority from the Master Mechanic or other official, except in case of accident, when notice must be sent at once to the Master Mechanic.
2. A tag must be filled out and attached to the valve (inside or yard) every time a valve is closed.
3. Never shut off more sprinklers than necessary, and keep off only for shortest possible time.
4. Provide for immediate turning on of water in case of fire. Where large values are involved keep a man near the closed valve ready to open it on notice.
5. For long and important shut offs provide extra watchman to patrol shut off area and notify Insurance Companies.

The Master Mechanic or Superintendent should provide in his office a conspicuous board with hooks, for these danger signals, and with supply of extra cards. Preferably a white board with a red border marked at the top—
"NOTICE OF FIRE SPRINKLERS SHUT OFF."

This method of Coupon Danger Cards was invented and tested in the Remington Typewriter Works and appears to be the best yet devised. Its adoption everywhere is urged by the Associated Factory Mutual Fire Insurance Companies.

**A supply of these tags should be kept in Master Mechanic's Office, and may be obtained free of cost from the Associated Factory Mutual Fire Insurance Companies, Inspection Department, 31 Milk Street, Boston.**

**A few spare tags for emergency use should be kept conspicuously placed near desk of head of each department.**

Fig. 5a Reverse of Mutual Companies' Red Card

official, and then to see that the system is used. And it is safe to say that if an owner finds that this or some such system cannot be maintained in effective operation, it may be taken as a symptom of the need of better organization in that factory.

## DISCUSSION

George I. Rockwood. As Mr. Miller points out in his very interesting paper, the possibility of the unauthorized closure of automatic sprinkler system shut-off valves is the only remaining nightmare before the eyes of fire protection experts. In view of the record of fire losses due to such unauthorized or accidental shutting off of the water supply to sprinkler systems, it is singular that so little has been done to give any *automatic* protection against such a possibility. All the rest of the apparatus, such as the sprinkler head, the alarm valve, the dry pipe valve, and the system of sprinkler piping, has been developed to a high pitch of perfection; but this one point remains to be developed.

Impressed with the importance of this problem, I have invented and developed a simple and inexpensive piece of mechanical and electrical apparatus designed to supplement the very suggestions contained in Mr. Miller's paper. Mr. Miller desires the master mechanic, or the person in charge of the sprinkler systems in an establishment, to divide a printed card in the middle, fill out on each half the reason for closing the shut-off valve and the time when it may be expected to be opened and the pressure restored to the system again, and then requires him to tie one-half of the broken card to the stem of the gate valve and to carry the other half into the office and leave it with some authorized person who shall share with him the responsibility for shutting off the water supply.

This system would be, of course, entirely satisfactory provided the person who is required to carry it out really does so. It does not take very much imagination, however, to suggest many everyday reasons why it would be for the interests of such a person to omit such notice altogether, in which event there is a large possibility that the valve may be left shut, whether intentionally or inadvertently, for a long period of time. For example, a workman may go home to luncheon at noon leaving the valve shut, and something may interfere with his return; if the job is to be done at night, or the job of repairing the sprinkler system extends into the evening, the person notified by the other half of the card may be absent until the next day, in which case

it might easily be that the valve would be left shut at least until his return.

The device which I have developed makes it *necessary* for the master mechanic to attach his card directly to an electric switch that controls an electric bell, which switch must be opened before he can attach it, and cannot be left open without the aid of a second person. This second person should be the one relied upon to see that the valve is actually reopened at the appointed time. He is also the one who would pay attention in case the valve was shut by some unauthorized person who had not first gone through the process of notifying the man in the office, and getting his assistance in holding the switch open, as this would result in turning in an alarm directly in the office. The way this is effected is by attaching an electric circuit closer to the valve stem of the main shut-off valve in a way such that any motion of the valve stem in the direction of shutting the valve will immediately cause the circuit closer to close the electric circuit through the alarm bell and the red light in the office.

When the switch is opened through an arc of 180 deg. it will be held open provided the master mechanic has caused the circuit closer on the valve stem to close by shutting the valve; but this would result in an alarm in the office if the assistance of the man who is to be notified that the valve is to be shut has not actually been obtained to hold the controlling switch clear open against an electric magnet energized by the same current that lights the light and rings the bell, while the other man goes to the sprinkler riser and shuts the valve.

It will be observed that this mechanical and electric supervisory system is entirely supplementary to the system proposed by Mr. Miller, that it does not interfere with it in the slightest degree, and that its influence is positive if it actually works as intended, but that if it should fail to work there would be no interference with the operation of the simple card scheme. This system has been in operation in my own factory for several months, has been demonstrated to various underwriters, and seems to be quite satisfactory. No doubt about its operation, either theoretically or actually, has thus far developed.

The principle covering the development of this device is that more than one man should have something to say about the opening and shutting of sprinkler valves, and that the operations of the extra person should be mechanically interlocked with those of the person directly charged with the oversight of the sprinkler system. The device in question, in addition to being supervisory over the motions

of the shut-off valves, is also a fire alarm and a water pressure alarm. In regard to the latter point, it may be said that the bell will ring and the light will light if the water pressure falls beyond a certain predetermined point.

In the history of sprinkler systems it has frequently happened that main shut-off valves have been shut with a heavy hand and then *apparently* opened again, when, in reality, it was subsequently found out that the only part of the valve which opened was the stem, which had become detached from the plug. This may easily happen when the valve stem is shut with a big wrench in an outside gate and the stem is twisted after the valve has bedded upon its seat. In such a case as that, my alarm system would call the fact to the attention of the office automatically.

James P. Tolman. It appears to me that Mr. Miller's instruction tag does not afford all the protection that is possible. In our mill we have used a tag similar in style to that, but in place of the circular hole we have a slotted hole, through which is passed a piece of lace leather. The leather is also passed through the yoke and the arm of the hand wheel, which seals the valve. The lace leather is secured by a tubular rivet, easily applied. Each valve which controls sprinklers throughout the mill is always sealed, either open or shut. If it is a valve which should be closed in winter, it is sealed shut. If it is a valve which should be open at all times, it is always sealed open, and the tag requires any person breaking the seal to take the tag to the office. Recently the Mutual Insurance Companies have been sending out the tag shown in the latter part of the paper, and we felt that it was an improvement and adopted it. But we also retain our old form of tag, which we now make from green card, and this green tag is on every normally conditioned valve at all times, showing that the valve is sealed, and that persons who break the seal of the valve are to take the tag immediately to the office. If a valve has its seal broken, there is a notice. The red tag has supplanted the green tag, and every valve should always have one or the other tag upon it.

One of the objections made by Mr. Rockwood to the tag is that when a valve was marked with a red tag, it would never be seen by anybody unless he *happened* to see it. I want to say that it is the practice of all the mutually insured mills to have weekly inspections of the fire protective system. We have different men inspect the system for different periods of time. If a valve is wrongly marked,

having the red tag instead of a green tag, it will be brought to the attention of the superintendent at least as soon as Monday morning.

Gorham Dana.[1] I agree with the previous speakers that the matter of a closed valve in the sprinkler system is a most vital point in the matter of fire protection in sprinkler-equipped plants. If we could eliminate the unnecessarily closed valve, the fire loss would be very materially reduced.

The scheme which one of the speakers proposed is an admirable one for certain conditions, but it does not seem to cover our conditions. In order to succeed with such a plan, it is necessary that a plant should be extremely well managed, with an efficient master mechanic and an efficient superintendent. There are a good many plants equipped with automatic sprinklers that do not have these conditions, a great many tenant factories, and a great many New York loft buildings, where there are many different firms in the building, and where it is difficult to get at any one to care for the sprinkler system. There are a great many small plants in various towns and cities that have no master mechanic who is worthy of the name, where the problem is very different.

The organization with which I am connected has been working on this problem for a number of years, and we have solved it in rather a different manner, which, to my mind, covers cases which this scheme would not cover. The plan is to have the valve fastened by the insurance inspectors. The tag which is used (Fig. 6) is slipped over the seal and is used in case the seal is broken to notify the insurance company. That is to say, the tag is similar to the one shown here, but somewhat differently worded, and in case the seal has to be broken to close the valve, the person who breaks the seal fills out certain data as to the circumstances (the reason for closing the valve, etc., the present condition of the valve, whether it was opened at once or not) and this information is put on the back of the tag and the tag mailed to the insurance inspection department. We find it to be extremely necessary in any plant which has not an efficient management, to have this information sent to the insurance interests.

Mr. Rockwood's scheme is a very excellent one, and I trust it will come into general use, but it has, however, one drawback: if a valve is closed for any length of time the alarm does not do much good. If

[1] Manager, The Underwriters' Bureau of New England, 141 Milk Street, Boston, Mass.

Name of risk.................... File No....................

TRIP NO. SPRINKLER VALVE

CONTROLLING.................... Valve No....................

Sealed (date).................... by ....................

In case seal has to be broken please fill in the following data, TEAR OFF THIS TAG WHERE COUPONED and send this tag at once to address on reverse side. Open the valve again as soon as possible, strapping it with leather strap-ends riveted or padlocked.

This seal IN NO WAY RELIEVES THE MANAGEMENT OF THIS PLANT OF FULL RESPONSIBILITY FOR PROPER CARE OF VALVES and other fire appliances.

Seal broken (date)....................

Cause....................

Is Valve still shut.................... Signed....................

TEAR OFF HERE.

FIG. 6 FORM OF REPORT TAG IN USE BY UNDERWRITERS' BUREAU OF NEW ENGLAND

a valve is closed for repairs, for an hour or two hours, you have got to shut off the alarm and cannot have it ringing all the time, and for that reason some notification to the people most vitally interested, the insurance companies, is most desirable in my opinion.

W. H. KENERSON (written). There can be no difference of opinion in regard to the necessity for keeping all sprinkler valves open, and if it is impossible to provide some of the mechanical supervisory devices, the scheme outlined in the paper is highly desirable. I take it that the purpose of the paper it to put before the public a scheme which anybody can employ, and the description of the system is in such detail that I beg to suggest one further simple expedient which will make it more effective. The majority of us have probably had difficulty, when using a system employing tags hanging on open hooks, in keeping the tags in place because of drafts, careless handling, etc. The simple expedient of turning the hooks upside down, will almost entirely obviate the difficulty. Such an arrangement would make this system much more reliable.

CHARLES H. BIGELOW. While I believe the tag is a good thing, exceptional discipline is required to keep it up in every case, especially in places having only a small mechanical force of the class often employed, and where the office is a long distance from the valves. There the tendency would be, especially for a short job, for the man to shut off the valve and do the work, instead of taking the tag to the office, returning to do the work and then going back to the office after the tag, particularly as many of them regard such regulations as "red tape" if they think of them at all. The trouble is, however, that while the men may open up the valve in most cases, there is always the chance of the man being called away and forgetting to open the valve, so that probably the tag system, even if not kept up the way it should be, would be a help.

POMEROY W. POWER (written). The weekly inspection seems to be a very good system to use in connection with any method of keeping track of sprinkler valves. Our plan is giving good results. Every Thursday an inspector examines each indicator post gate and signs a tag, which is attached to the post, with his name and the date. After his examination of all sprinkler valves, the inspector signs a typewritten form of letter, which reads something like this: "I have this day inspected all of the sprinkler valves. They are all open and

TABLE 1 INSPECTION RECORD OF HOSE HOUSES AND SPRINKLER VALVES

Andrew McLean Co., Passaic, N. J.

| | Location | Sizes In. | State of Valve |
|---|---|---|---|
| 1 | Third St., gate valve, city connection | 6 | Open |
| 2 | Mill No. 1, napping depart., indicator post | 6 | Open |
| 3 | Mill No. 1, napping department, drip | 2 | Closed |
| 4 | Hose house No. 1, two-way hydrant, ft. hose washers, spanners, nozzle. | | Clear |
| 5 | Boiler house, sprinkler line | 3 | Open |
| 6 | Boiler house, sprinkler drip | 1½ | Closed |
| 7 | Boiler house, 6-in. gate in line to mill No. 1 | 6 | Open |
| 8 | Old boiler house, drip | 2½ | Closed |
| 9 | Old boiler house, sprinklers under gallery | 2 | Open |
| 10 | Kier room, kier dye and breach room sprinklers | 3½ | Open |
| 11 | Kier room, drip for above | 1¼ | Closed |
| 12 | Old grey room, drip | 2 | Closed |
| 13 | Mill No. 1, second floor sprinklers for tower | 1¼ | Open |
| 14 | Mill No. 1, indicator post (tower entrance) | 6 | Open |
| 15 | Mill No. 1, indicator post (shipping room) | 6 | Open |
| 16 | Drip in shipping room | 1¼ | Closed |
| 17 | Hose house No. 2, one-way hydrant, ft. hose washers, spanners, nozzle. | | Clear |
| 18 | Storehouse No. 2, indicator post | 6 | Open |
| 19 | Storehouse No. 2, basement drip | 2 | Closed |
| 20 | Office, indicator post | 6 | Open |
| 21 | Office, drip in pit of 1st toilet | 2 | Closed |
| 22 | 6-in. valve in main rear shop | 6 | Open |
| 23 | Storehouse No. 3, basement drip | 2 | Closed |
| 24 | Storehouse No. 3, two-story end, indicator post | 6 | Open |
| 25 | Storehouse No. 3, two-story end, drip | 2 | Closed |
| 26 | Storehouse No. 3, one-story end, indicator post | 6 | Open |
| 27 | Hose house No. 3, two-way hydrant, ft. hose washers, spanners, nozzle. | | Clear |
| 28 | Hose house No. 4, two-way hydrant, ft. hose washers, spanners, nozzle. | | Clear |
| 29 | Hose house No. 5, two-way hydrant, ft. hose washers, spanners, nozzle. | | Clear |
| 30 | Mill No. 2, indicator post | 6 | Open |
| 31 | Mill No. 2, drip | 2½ | Closed |
| 32 | Hose house No. 6, two-way hydrant, ft. hose spanners, washers, nozzle. | | Clear |

The above has been examined by me to-day.

Date, .................... Signed, ........................................

Countersigned by ..................................................

in good order, except——" Under the list of exemptions he fills in the location of any valve that is closed, the date when it was closed and the date that it is expected to be opened. That letter, after being signed by the inspector, is sent to the manager as a weekly report. After about four years use of this plan, we have had only one case of a valve being found closed.

Lewis H. Kunhardt. This whole question comes down to a matter of responsibility. Some one must be made absolutely responsible in a plant for the maintenance, for instance, of the sprinkler valves, the same as some one must be made responsible for the maintenance of any other piece of apparatus in the plant. A combination of the inspection and the tag will, we believe, fulfill these conditions of keeping the valves open. Of course, there is a fair possibility that something may slip up, but when it does slip up, the point is that you must have some responsible person to whom to look to see why it occurs, and that it does not occur again. Any one who tries this combined plan, that is, a system of inspection and tagging the valves, will accomplish the result.

F. J. Bryant (written). At our plant we have inspection cards giving in itemized detail the route to be followed, showing the location, size, type and duty of each valve, the location of each length of hose, each nozzle, tool, etc., as shown in Table 1. The man making the inspection takes the card and checks each item as he comes to it (if he finds it correct) and notes any discrepancies on the margin. After completing the round he signs the card, as the previous speaker said, and places it upon the manager's desk, who countersigns it and has it filed for further reference.

The Author. Before adopting this system of cards, we carefully considered mechanical and electrical devices for controlling our valves. There are men in our organization who have devoted a great deal of study to electrical devices and they at once thought of like means of controlling valve opening and closing, and proposed several schemes. We have, after careful study of the matter, come to the conclusion that it is better to depend upon our discipline and organization and to have some one responsible for these valves. If we put a mechanical or electrical device on the valves, any one would still be able to close it at will, and it would not involve the doing of anything else. We believed that notwithstanding the use of such a device we would have

to go to some form of tag and report in addition. It is a fact that any automatic device, such as a water level indicator in a boiler, or anything of that sort, that is depended upon to indicate danger or abnormal conditions is apt to be depended upon too much; some day it will not work and then there is disaster. For our conditions at any rate, a thoroughly organized system for taking care of these valves and a definite placing of responsibility upon someone for their condition is the best. The seal idea for the card seems to be good, and would be a useful addition to it.

We had, as stated in the paper, before this card system was installed, a very thorough system of inspection for these valves, and we still maintain that inspection system. But the trouble about a weekly inspection is that if the inspector looks at a valve on a Thursday morning, the valve may be closed until the next Thursday morning and no one know anything about it except the man who closed it. Our system provides that someone must know about it; some responsible person must know immediately if the valve is closed, and we think that is important. Of course, there must be discipline, a definite placing of responsibility and strict accountability.

## No. 1423

# THE NEED OF MORE CARE IN THE DESIGN AND CONSTRUCTION OF ELEVATED TANKS

By W. O. Teague, Boston, Mass.

Member of the Society

The elevated or gravity tank for fire protection systems has been from the first an important limited secondary source of water supply, and its value has increased greatly with the increase in number and size of tanks installed generally throughout the country, especially in those cities and districts where the public water supply is of low pressure, as is the case in Philadelphia. The tanks are usually located above buildings in cities and on detached towers in the country. A typical detached structure is shown in Fig. 1, which is a 150,000-gal. tank on a 242-ft. tower at the plant of the New York Shipbuilding Company, Camden, N. J.

2 The tanks were first made of wood, but there are now as many being made of steel. Wooden tanks have been built up to 100,000 gal. capacity, although they are rarely larger than 60,000 gal., for above this capacity the steel tank is cheaper and more practicable. The cost of a 60,000-gal. tank of wood or steel erected on a 75-ft. steel tower is about $3,000. Steel tanks are built in large sizes, one of the largest being of 1,200,000 gal. capacity; this one is 50 ft. in diameter and 90 ft. high, and is supported by a steel tower 130 ft. high.

3 Failures of tanks in service, involving loss of life and destruction of property, have shown the need of more care in the designing and construction of them. To insure the best results, the following features should have attention.

### WOODEN TANKS

4 The tightness and durability in the wooden tank depends chiefly upon the quality of the lumber and the details of its construction. Selected tank stock only should be used consisting of white cedar, cypress, white pine, Douglas or Washington fir, or redwood, and the lumber should be free from sap, loose or unsound knots, worm-holes

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

and shakes, and be thoroughly air-dried. Both the staves and bottom are usually made up of 2½-in. stock dressed both sides, for tanks up to 16 ft. in diameter and 16 ft. deep; for larger tanks 3-in. stock is used. Planks for this purpose should be full length without splices.

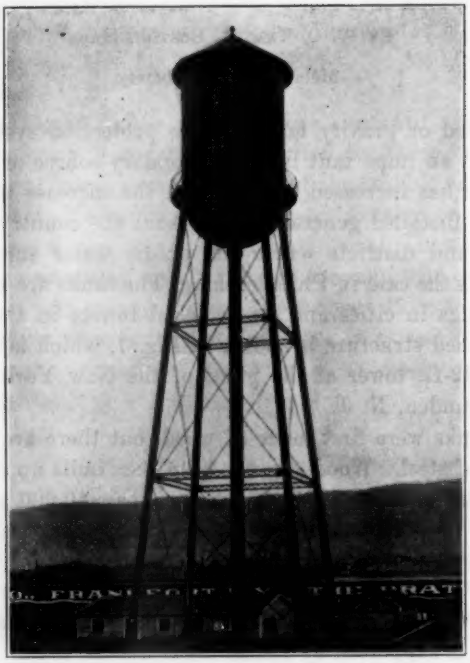


FIG. 1 TYPICAL DETACHED TOWER STRUCTURE FOR ELEVATED TANK. CAPACITY 150,000 GAL.; ELEVATION 242 FT. TO BOTTOM OF TANK

5 The strength of the wooden tank depends principally upon the size and spacing of the iron hoops. The importance of the matter of the hooping will be appreciated when it is realized that the overstressing of even one hoop may result in the bursting of the tank. The wooden tank being originally merely a development of the barrel where flat hoops were necessary to permit of tightening by driving them toward the enlarged middle, it was natural to use also flat hoops for the tank and the tank was also made tapered so that the hoops could be tightened by driving, although later they were tightened principally by hoop lugs. It was claimed that the tapered shape had also the advan-

tage of preventing the hoops from dropping down over the tank, if it was allowed to remain empty and the staves to shrink from drying.

6 The tapered shape of tank is not important, however, since a tank which has been allowed to dry up, has been seriously damaged thereby and cannot be made tight without extensive repairs, sometimes necessitating the rebuilding of it. In fact most tanks are now made without taper and the hoops are found to remain where placed. The tapered tank costs somewhat more to build since the staves must be fitted more carefully and the design undoubtedly would have been entirely discarded long ago, except that some architects and purchasers believe a tapered tank presents a more pleasing appearance. The amount of taper is so small, being usually 1 in. per ft., thus giving a batter of ½ in. per ft. to each side of tank, that its absence is hardly noticeable except on very high and small diameter tanks. The only objection to the tapered tank, however, is its extra cost.

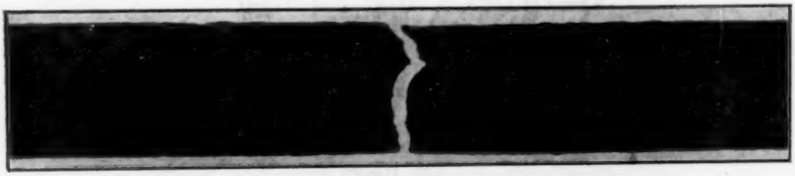


Fig. 2 Appearance of Inside Surface of Flat Band Hoop that Rusted through and Failed on Wooden Tank

7 In the early studies of this subject many serious failures of tanks were traced to weakening of the flat hoops by their rusting at the back where they bore against the staves, due to moisture from rain being retained between the surfaces of the hoop and staves. Fig. 2 shows the appearance of the inside face of a flat hoop which rusted through and allowed a tank to burst. These failures were largely unpreventable, as it was difficult to inspect properly the condition of the hoops, and also impossible to paint them while the tank was in service. The use of hoops of round rod without welds has remedied this trouble as their surface is nearly all exposed for inspection and painting, and also they are not so subject to corrosion since the exposed surface of a round rod is much less than that of a flat bar or band of the same cross sectional area.

8 Round rod hoops are now being used exclusively on well built sprinkler tanks, and the flat hoops of a large percentage of previously built tanks have been replaced with them. It is, of course, advisable

that the remaining flat hoops also be replaced, but if this is not done, the hoops should be carefully examined every few years to note their condition. In making such an examination, a round rod hoop should be placed on each side of one of the lower flat ones, within easy reach from the balcony; the flat hoop should then be struck smartly with a pointed hammer at intervals of a few inches all around to detect thin places, or else the hoop should be removed and examined. Examination at wider intervals is not sufficient since the hoop may be in good condition at one point and be nearly rusted off at a point six inches away. If this lower flat hoop is found to be corroded materially, all the hoops should be replaced with round rod ones, or, if preferred, the flat ones may be retained and round rod ones placed between them.

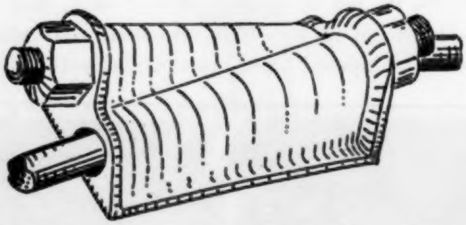


Fig. 3 Preferable Design for End Lugs in Malleable Iron for Round Rod Hoops

9 Another point of weakness in the flat hoop is at its connection to the cast-iron lugs which is usually made by riveting. The use of round rod hoops, however, permits of a satisfactory connection to the lugs, but at first many tank failures resulted from the use of light cast iron lugs. These are now made of malleable iron, the best design being shown in Fig. 3. The hoops are so placed on the tank that the lugs do not come in a vertical line.

10 Round rod hoops are so spaced that the stress will not exceed 12,500 lb. per sq. in. when computed from area at the root of the thread. The proper spacing can readily be found from the following formula:

$$\text{Spacing of hoops (in.)} = \frac{\text{Safe load for given hoops (lb.)}}{2.6 \times \text{diameter (ft.)} \times \text{depth (ft.)}}$$

The depth used is the distance from overflow to point where hoop is to be located. The top hoop is placed 2 in. from the top of the staves and the spacing between hoops should in no case exceed 21 in. An extra hoop or two is placed at the croze to take the additional strain due to the swelling of the bottom planks.

11 The tank roof, since it in no way serves to retain the water, has usually been nothing more than a make-shift cover. In the early days a single flat roof was used on outdoor tanks, but this held the snow and ice and required strong joist supports to keep it tightly in place. The snow also interfered seriously with the opening of the hatch giving access to the interior of the tank. A conical roof was then built over the flat one which remedied these difficulties. It also greatly increases the efficiency of the roof in preventing radiation of heat from the tank water in winter as it provides a dead air space between it and the flat roof in addition to the one between the latter and the water, thus reducing the cost of heating in freezing weather. The conical roof also gives a better appearance to the tank top. A

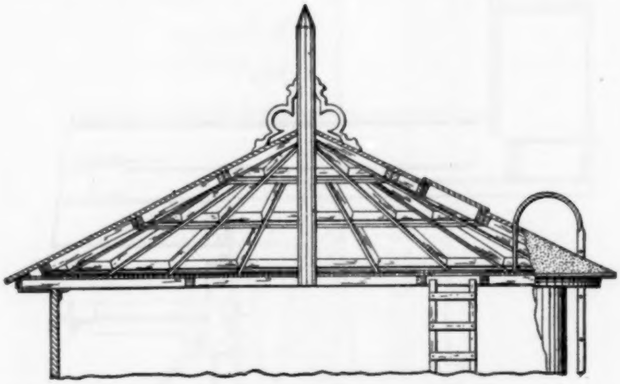


Fig. 4 A Good Design of Double Roof for Elevated Tank

well built roof is shown in Fig. 4. It should be tightly fitted around the tank top to maintain the dead air spaces.

12 The roof covering which was most generally used was the wooden shingle, but these were found to catch fire from burning embers which spread to the tank staves, weakening them so that the tank burst. Non-combustible coverings, such as galvanized sheet iron and composition roofings are now used.

13 Much trouble has resulted from leakage in the wooden tank, because it has not been firmly supported. The wooden tank is locally weak, not being of unit construction, and the lack of firm support has permitted working of the joints. It is supported only from the bottom, none of the weight being carried by the staves. Wooden beams were first used as supporting members, and these were placed on the

roof of a building or tower as a grillage, and the tank bottom set on them. In time the wood rotted because of moisture from the tank bottom, permitting the tank to settle and causing leakage; there was also danger of collapse of the tank because of this weakening of the joints. The use of steel I-beams as grillage members as shown in Fig. 5 avoids these difficulties. The beams should not be spaced over 18 inches clear between edges of flanges, and the tank bottom is placed directly on the steel.

### STEEL TANKS

14 When good tank lumber began to get scarce and to increase

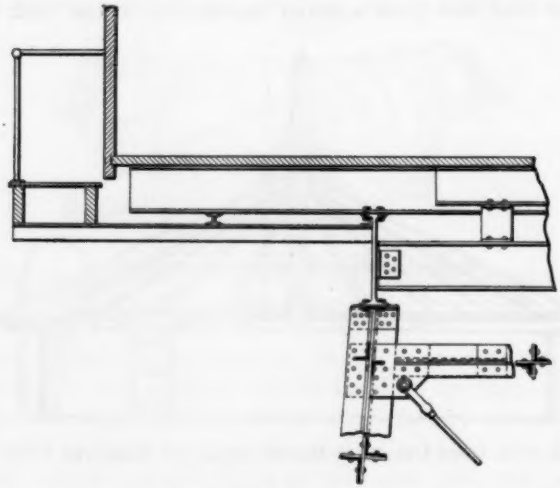


FIG. 5 DETAIL VIEW OF ARRANGEMENT OF STEEL I-BEAM GRILLAGE FOR SUPPORT OF TANK

considerably in price, boiler and iron companies turned their attention to the manufacture of steel tanks and towers, especially of the larger sizes. In the design and construction of these tanks, the manufacturers have drawn on their extended and complete experience in boiler and bridge work, but the matter of supporting the tank properly had to be experimented with until experience with tanks in service was available. The simplest form of steel tank is the flat bottomed one and tanks of this type give satisfactory service, provided the bottom is supported by a steel grillage as in the case of the wooden tank. One possible source of trouble is from corrosion of the bottom, and to prevent this in so far as possible the bottom plates are made

somewhat thicker than is necessary for strength alone, and the grillage I-beams are of a height and spacing to permit of inspection and painting of the bottom. When the tank is to be placed on a concrete tower, it may rest directly on a reinforced-concrete slab with the bottom thoroughly grouted in place with neat cement.

15 The preferred form of a tank to be placed on a steel tower is that having a hemispherical or elliptical bottom. The construction in this form is cheaper than for the flat bottomed tank as the bottom is self supporting and a steel grillage is unnecessary. The entire bottom is also accessible for inspection and painting, and corrosion is reduced to a minimum since the plates are exposed to the air.

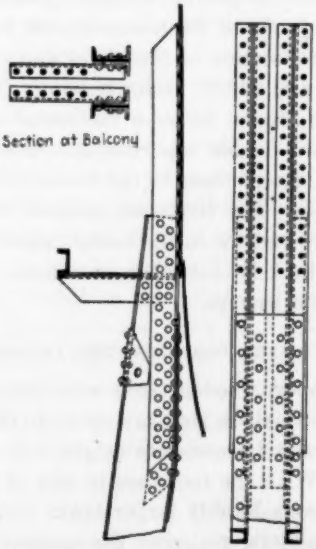



FIG. 6 DETAIL OF ATTACHMENT OF TANK SHELL TO TOWER POST AND CIRCULAR GIRDER CONSTRUCTION

16 Plates for use in steel tanks are made somewhat thicker than is necessary for strength in order to make them durable against corrosion. The minimum thickness is ¼ in., except that ⅛-in. plates are used for roofs. The plates composing the lowest cylindrical ring are 5/16 in. thick for 60,000-gal. tanks and larger, and the bottom plates 5/16 in. thick for tanks 75,000 gal. and larger.

17 One of the weaknesses of steel tank construction in the past has been poorly designed connections of the tank shell to the posts of the supporting tower. When the posts have a batter, as is usually the

case, the inward thrust due to the horizontal component of the weight is provided for by a circular girder consisting of ¼-in. plate 24 in. wide, attached to the lowest cylindrical ring by an angle and stiffened by angles or a channel at the outside edge, as shown in Fig. 6. The posts also connect to the tank shell at this point and the design is such that the load will be transferred from the shell to the center line of the posts so as to avoid eccentric loading. A number of tanks built without circular girders have failed by the posts crushing in the tank plates. Others with the girder, but having eccentrically loaded connections to the posts, have failed by bending of the upper posts.

18 As the hydrostatic pressure on the tanks is comparatively small, it is not necessary to provide standard riveting for the thickness of plates used. The joints of the plates should be riveted so that the unit stresses on the net section of the plates and rivets will not exceed 7500 lb. for shearing and 20,000 lb. for bearing. The horizontal joints are single lap riveted, except between the lowest cylindrical ring and the bottom, which are double lap riveted. The vertical joints also are single lap riveted except those in the lowest cylindrical ring, which are double lap riveted. The rivets are entered from the outside and driven from the inside and the inside seams calked. One of the strong features of the steel tank is that when once made tight, it gives practically no trouble from leakage.

## TOWERS FOR ELEVATED TANKS

19 Towers to support wooden tanks were originally built of wood. While the tanks were small in size, say up to 20,000 gal. capacity, and elevated to a comparatively moderate height, this construction proved fairly satisfactory. With the increases in size of plant buildings and extensions of them, considerably larger tanks and higher towers were required, and the builders, realizing the inadequacy of wooden construction under these conditions, began to make towers of steel. One manufacturer built towers of iron pipe construction having from four to twelve posts and the members connected by cast iron fittings, but it was practically impossible to provide resistance against uplift due to wind pressure, and with the tank empty the structure was in great danger of overturning. The inside of the pipe was subject to corrosion and as its condition could not be inspected or the surface painted, the metal might be greatly weakened without its showing on the outside. These towers were sold largely in a knocked-down condition by mail order and erected by the purchaser who of course could not be expected to have expert knowledge as to their construction. Ob-

viously, these towers should not be depended on for such an important use, and it is advisable to replace those already built with well-designed structural steel towers to forestall failure.

20 The wooden tank manufacturers' shops were originally fitted only with woodworking machinery, so that it became necessary for them to prepare for the fabrication of steel work. The managements not being experienced in structural steel designing, naturally selected the simplest design possible for the towers The posts and girts consisted usually of two angle irons, placed apex to apex and strapped together at intervals of several feet by tie-plates shop riveted to the angles. The column sections were spliced by angles which were shop-riveted at one end to the post; the other end was field-bolted in erecting the tower, as this was the simplest form of connection and the easiest one to make. Furthermore, it had the advantage that the bolting could be done by the regular erecters which made it unnecessary to have first-class mechanics in the erecting gangs and to carry special tools. This, however, was not good construction and the manufacturers are now field-riveting these connections.

21 The struts were at first connected directly to the posts by bolts. This construction is objectionable because the bolts are apt to work loose and it does not brace the parts. The construction now used is that of gusset plates riveted to the posts and girts. The wind rods were also connected directly to the posts at the girts. The bolt holes, as originally inserted through the post angles, weakened the posts since they reduced the net section. The rods are now connected to the gusset plates. The arrangement of these parts is shown in Fig. 5. The diameter of bolt and thickness of plate are proportioned to provide proper bearing strength.

22 The posts and girts of steel towers erected to support steel tanks, and to some extent wood tanks, are now largely made of channels latticed on both sides or having a plate on one side. Other shapes such as the Bethlehem H-beam and two channels with an I-beam between to form an H-section are also used to some extent.

23 Competition in the manufacture of these structures has resulted in the use of too high unit stresses and as a result the posts, figured on a conservative basis as represented in case of other structural work such as bridges, had a factor of safety of less than 4 and sometimes as low as $2\frac{1}{2}$. Failure has resulted, an example of which is shown in Fig. 7, a view of a large structure after a collapse. To obtain safe towers it became necessary, therefore, to set maximum allowable stresses. The loading of the structure consists of the weight of the

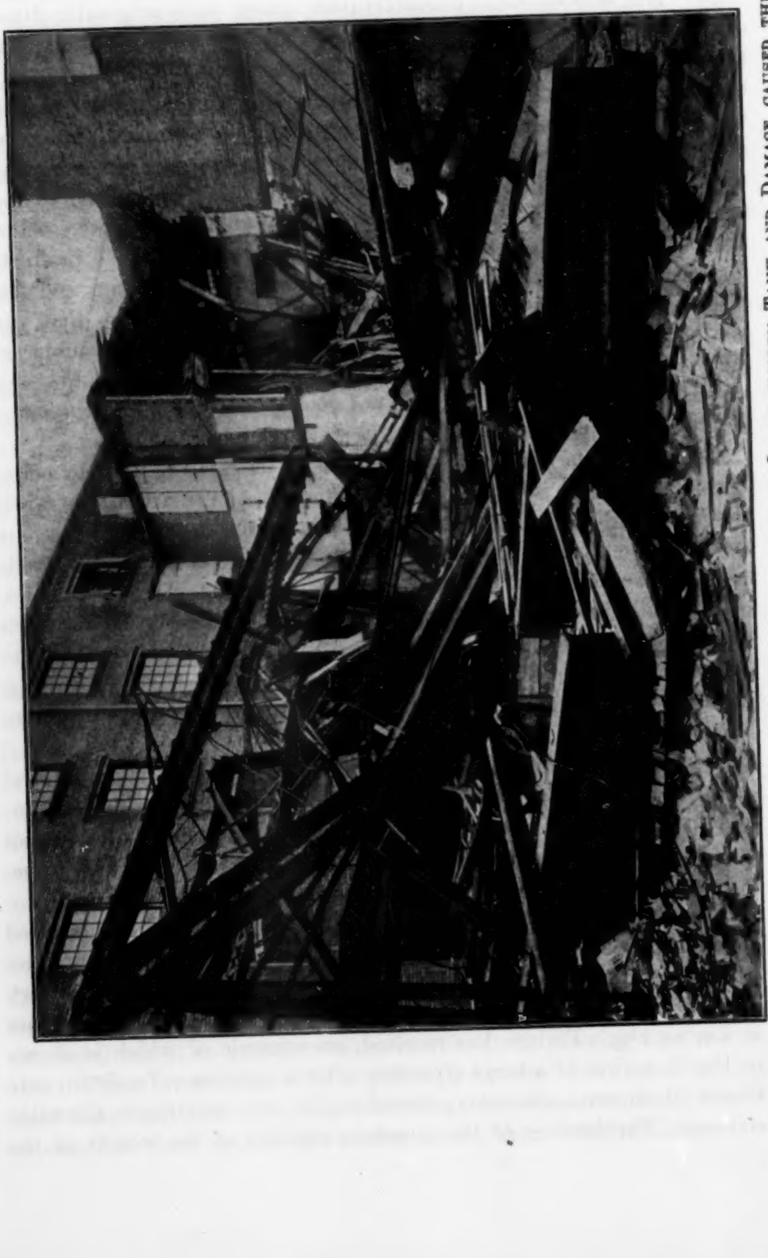


FIG. 7 VIEW OF COLLAPSE OF STEEL TOWER STRUCTURE WITH LARGE WOODEN TANK AND DAMAGE CAUSED THEREBY

structural and ornamental steel work, platforms, roof, piping, etc. The live load consists of the weight of the total volume of water; the movable load on the platform is assumed to be 30 lb. per sq. ft. and the wind load. The wind pressure is assumed at 30 lb. per sq. ft. and that on the tank is this pressure times 6/10 the projected area of tank and roof, and in the case of steel tanks, the curved bottom. The total wind pressure on the posts, struts, wind rods, ladders and riser boxing is assumed at 200 lb. per linear ft. of height of tower.

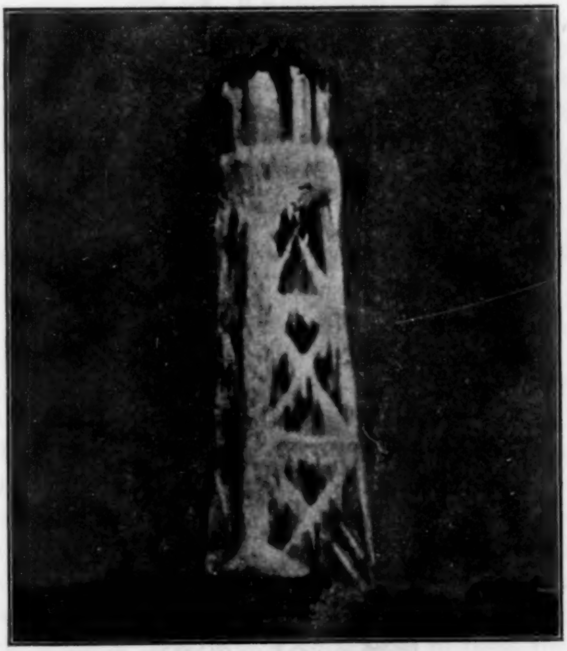


Fig. 8 Unusual Case of Loading of Large Steel Tower Structure by Ice; 60,000-Gal. Tank on a 79-Ft. Tower, uninjured after thaw

24 An unusual loading of a steel tower structure by ice is shown in Fig. 8. The tank is 60,000 gal. on 79-ft. tower and is located at Richfield, Idaho. The ice coating was formed from repeated overflowing of the tank during cold weather. When warm weather melted the ice, the structure was found to be uninjured. This experience illustrates the abuse a well-built structure will withstand.

25 All parts of the structure are proportioned so that the sum of the dead and live loads shall not cause the stresses to exceed those

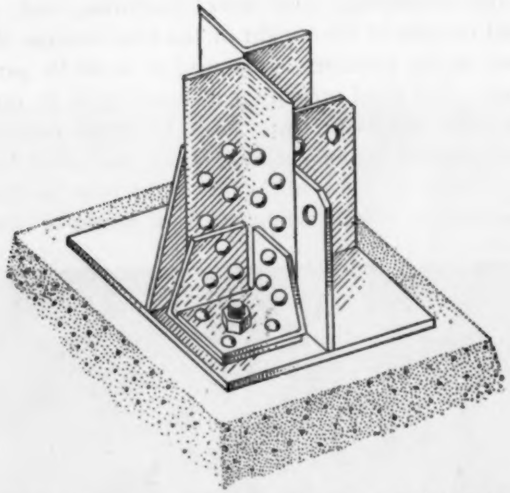


FIG. 9 TYPICAL CONSTRUCTION OF FOOTING FOR ANGLE IRON COLUMN

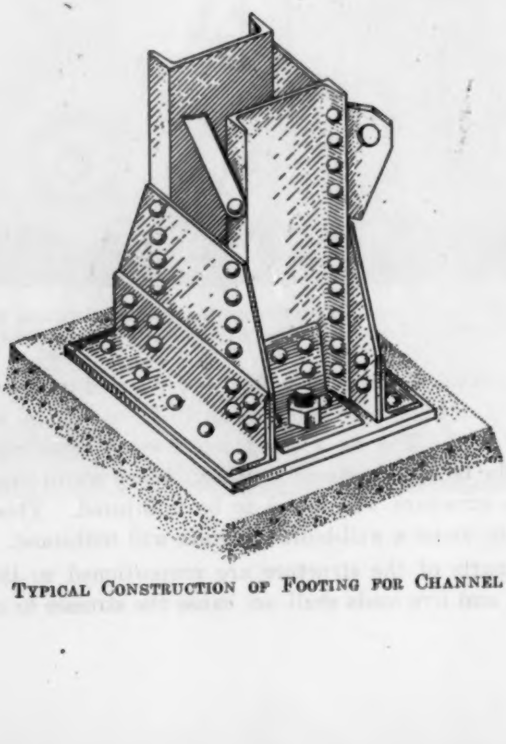


FIG. 10 TYPICAL CONSTRUCTION OF FOOTING FOR CHANNEL COLUMN

allowable. The principal stresses in such a tower structure are axial compression on gross section of columns and struts, axial tension on net section of wind rods, bending on extreme fibers or net section of rolled shapes, built sections and struts, and shearing of rivets. The axial compression on gross section of columns and struts is determined from the following expression: $17{,}100 - 57\frac{L}{R}$, where $L$ is the unsupported length of the member from center to center of connections in inches, and $R$ the least radius of gyration in inches; the ratio $\frac{L}{R}$ should not exceed 125 for columns and 150 for struts and minor members and the maximum compression allowable as thus determined is 12,000 lb. per sq. in. The axial tension on net section of wind rods must not exceed 12,500 lb. per sq. in.; the bending on extreme fibers or net section of rolled shapes, built sections and struts 16,000 lb. per sq. in., and the shearing for shop driven rivets, 10,000 lb. per sq. in. and field driven rivets 7,500 lb. per sq. in.

26 The lower ends of the posts have not been as carefully designed as their importance requires. Frequently, in angle iron towers particularly, no special attempt has been made to properly distribute the load to the base plate attached to the post footing. Cast-iron plates were first used and the concentrated loading caused these to crack, resulting in collapse or in throwing the structure dangerously out of plumb with possibility of failure of the foundation under this post. The present use of steel plates has improved conditions, but the design must be such as to distribute the load to them as shown in Figs. 9 and 10, which are designs that are being used quite generally.

27 In anchoring the columns to the foundations, the diameter of the bolt at root of thread should be such as to withstand the maximum uplift due to the wind with tank empty, and to resist the shearing force at base plate. The bolts should be made from round wrought iron or mild steel rods without upsets.

## FOUNDATIONS AND SUPPORTS

28 The foundation piers to support steel towers are usually made of concrete, consisting of one part portland cement, three parts clean sand and five parts broken stone. They are usually pyramidical in shape and proportioned to suit soil conditions. The allowable bearing pressures on soil will range from 1 to 5 tons per sq. ft., depending on the quality of the soil. Where the soil is moist or rather loose, a

girt should be provided at the base of the tower to prevent spreading of the posts. The allowable bearing pressures for footings should not exceed 400 lb. per sq. in. for portland cement concrete and 200 lb. per sq. in. for ordinary brick work with portland cement mortar, except when the tank is to be rested on the building walls, when the bearing plate should be figured on the basis of 125 lb. per sq. in.

29 The weight of the foundation pier when buried at least two-thirds of its height should be equivalent to the calculated net uplift due to wind pressure with the tank empty, that will be transmitted to it; otherwise it should be one and one-half times that amount.

30 Where the tank structure is above a building, and the building walls are depended upon to act as supports, great care should be taken to determine that the construction is safe against collapse. In many cases, tanks are supported by building walls not originally built to carry them, but where a sprinkler system was later installed it was considered more convenient and cheaper to use the walls than to erect a detached tower for the tank. This has frequently been done without making a thorough inspection first of the condition of the walls, and, largely through ignorance, the necessary care was not taken to distribute the load. Many failures have consequently resulted and there are no doubt numerous cases of this kind where the tanks are apt to fall at any time. A serious cracking of the walls in such an installation is shown in Fig. 11.

31 Inspection should be made of the quality and condition of the brick and mortar or other material used in the construction. The wall foundations should be examined as to construction and bearing on soil or rock. The condition of the bond between abutting walls should be noted and a general inspection made for sizable cracks in the walls. The thickness of walls and size and spacing of window and door openings should be measured and calculations then made to determine if the load of tank, water and trestle can be safely distributed over the walls. All unnecessary openings should be bricked or otherwise solidly filled in, and it may be necessary to sacrifice some openings to obtain the required strength. When the walls cannot be altered to support the load, the additional support required can be obtained by carrying steel beams down inside the walls to a solid foundation, provided these do not interfere with the occupation of or the processes carried on in the building. Otherwise it will be necessary to provide a separate steel tower.

32 The proper strength of foundations is especially important because of the greater probability of loss of life from the falling of a tank from above a building as compared with the falling of a tank on a detached tower. The monetary loss is liable, of course, to be also much greater, as the water will undoubtedly wreck the building and cause heavy water damage. The building departments of cities endeavor to obtain proper construction, but unfortunately they do not

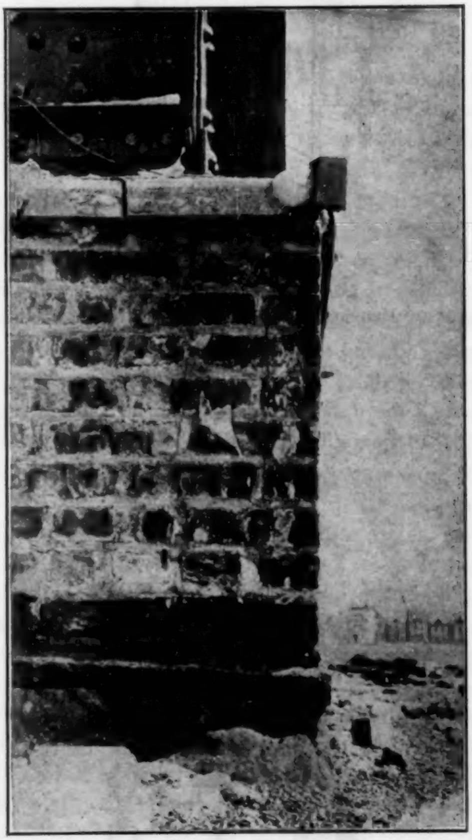


Fig. 11 View of Supporting Wall that has Cracked under Weight of Tank

always succeed. The possibility of trouble is increased because of the divided responsibility of the tank builder and the architect. The former seldom concerns himself as to the strength of the supporting

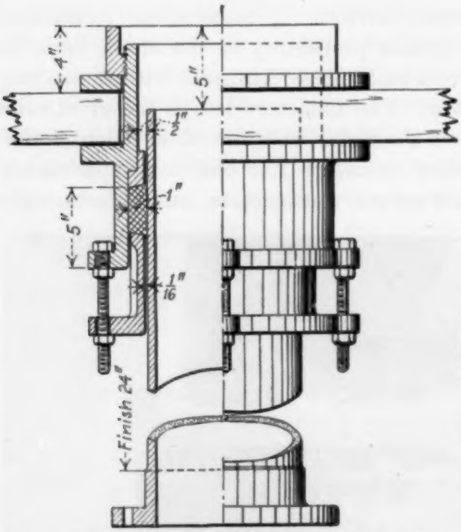



FIG. 12 EXAMPLE OF PROPER DESIGN OF EXPANSION JOINT FOR WOODEN TANK

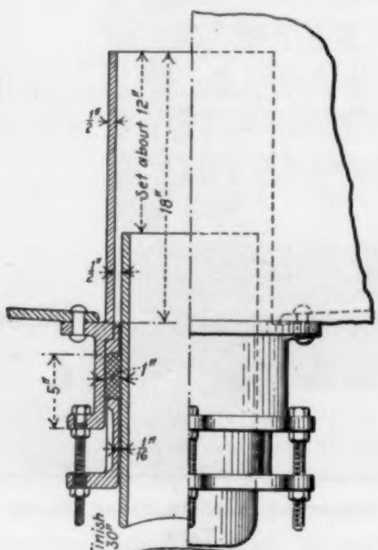



FIG. 13 EXAMPLE OF PROPER DESIGN OF EXPANSION JOINT FOR STEEL TANK

walls, assuming that the latter has given the matter proper attention, so he goes ahead and erects the tank according to contract.

### GENERAL FEATURES

33 Tank fittings should receive careful attention to insure the reliability of the equipment. The discharge or riser pipe is more serviceable if made up of cast or wrought-iron pipe, flanged or coupled, than one made up of bell and spigot pipe, since the latter is apt to leak at the leaded connections, necessitating removal of the frost-proof boxing to permit of repairs. A tank and tower is constantly swaying from side to side and this tends to loosen up leaded joints. Furthermore, the increased rigidity of the flanged and coupled pipe permits the use of a minimum number of tie rods. There are usually four rods connected, one to each post, at girt connections.

34 The connection of the discharge or riser pipe to wooden tanks has usually been made by extending the pipe through ordinary cast-iron slip flanges bolted to the tank bottom on each side of the opening. The hole in the planks was cut larger than the size of the pipe to form a packing space which was filled when parts were first assembled. A better construction was used for steel tanks having a stuffing box and gland. Both types of joints were found to be unserviceable, however, the former because the joint could not be tightened when leakage occurred, and the latter principally because the iron to iron parts rusted together, which resulted in the breaking of some pipe fitting and the emptying of the tank. Examples of properly designed expansion joints forming tank connections for wooden and steel tanks are shown in Figs. 12 and 13, respectively. These have a bronze gland and ample clearance between the iron parts to prevent binding by corrosion. The packing space is large and the joint is extended within the tank bottom to form a settling basin, to prevent sediment getting into the yard pipe and clogging the sprinklers at time of fire.

35 A tightly constructed frost boxing should be placed around the discharge or riser pipe, and arrangements made for keeping the water heated by a hot water heater or a steam coil in the bottom of the tank. Designs of three-ply, two air-space boxings are shown in Figs. 14 and 15.

36 A tank level indicator or telltale is necessary to give a positive indication that the tank is full at all times. After many serious fires it has been learned that the tank had been partially or wholly empty at the start of the fire, and the lack of water had handicapped the fire protection devices. Tanks may be left empty due to neglect, but

usually so because of false indication of the telltale. The most used type of device for this purpose is the float in the tank water, operating a target sliding on a scale fixed to the outside of tank. Obviously,

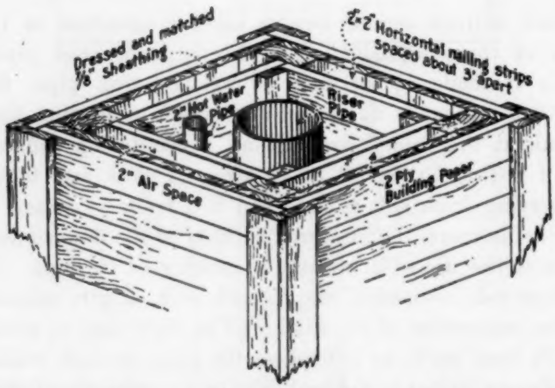



**Fig. 14 Design of Efficient Frost-Proof Square Boxing for Enclosure of Riser Pipe to Tank Connection**

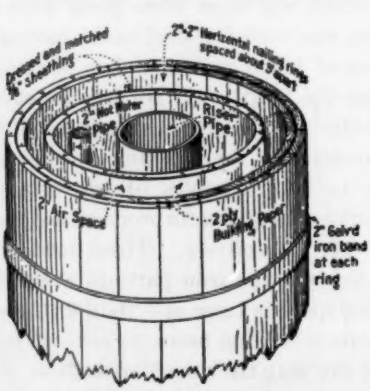



**Fig. 15 Design of Efficient Frost-Proof Round Boxing for Enclosure of Riser Pipe to Tank Connection**

these are subject to sticking due to their mechanical construction and exposure to snow and ice in freezing weather. The ordinary pressure gage has been largely used but cannot be positively depended on, since it is seldom, if ever, tested and the parts stick, causing false readings. There are several types of electrical telltales operated by a float, but these are complicated and easily gotten out of adjustment. Attention is also necessary to maintain the electrical current.

37 The most reliable telltale is undoubtedly the mercury gage, an adaptation of which for this purpose is shown in Fig. 16. This gage was developed by the laboratories of the Associated Factory Mutual Fire Insurance Companies. It should be placed indoors where it will be observed and cared for. The mercury pot is then piped to the

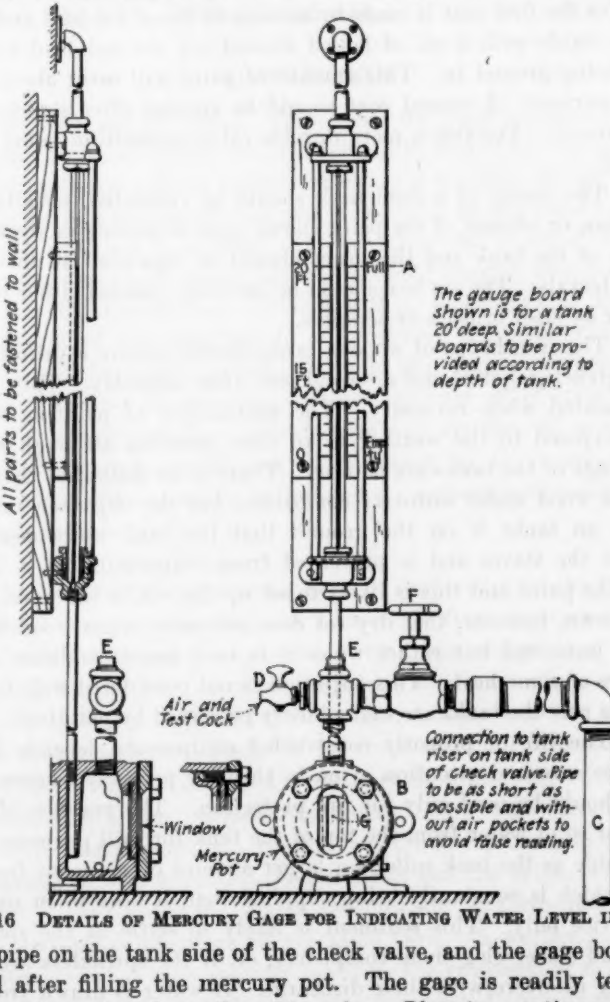



FIG. 16 DETAILS OF MERCURY GAGE FOR INDICATING WATER LEVEL IN TANK

riser pipe on the tank side of the check valve, and the gage board adjusted after filling the mercury pot. The gage is readily tested by opening the pet cock on the water pipe. If water continues to flow under constant pressure, the apparatus is in operative condition; otherwise, the pipe is clogged or there is a valve closed.

38 The painting of steel tanks and towers and of the iron hoops of wooden tanks is very important to prevent corrosion. Steel plates and shapes should be given the usual priming coat at the shop. The surface of the metal should be thoroughly cleaned of mill scale, rust and grease and be perfectly dry before applying the paint. A good paint for the first coat is made by mixing 20 lb. of red lead and 10 lb. of zinc oxide with 3 qt. of boiled linseed oil, the red lead and zinc oxide being ground in. This amount of paint will cover about 50 sq. yd. of surface. A second coat should be applied after structure has been erected. For this a more durable oil or asphaltum paint should be used.

39 The inside of a steel tank should be repainted, usually every two years, or oftener, if the paint shows signs of peeling or wear. The outside of the tank and the tower should be repainted at about five-year intervals. The surface should be carefully cleaned either by sand blast or by steel brushes or scrapers.

40 The iron hoops of wooden tanks should receive a priming coat as for structural steel and a second coat after assembly. They should be repainted when necessary. The advisability of painting wooden tanks exposed to the weather is an open question although a large percentage of the tanks are painted. There is no doubt but that paint protects wood under ordinary conditions, but the objection raised to its use on tanks is on the ground that the tank water percolates through the staves and is prevented from evaporating as it is held under the paint and this is likely to set up dry rot in the wood. It is well known, however, that dry rot does not occur when wood is completely immersed but rather when it is in a moist condition in the presence of some heat. This objection is not considered well-founded and as a rule the tanks are undoubtedly preserved by painting.

41 The life of properly constructed equipments depends largely upon the care and attention given to them by property owners. The tanks should be used only for fire protection. The practice of using a foot or so of water from the top of the tank for mill purposes is objectionable as the tank collects a larger amount of sediment from the water which is constantly being supplied than it does when used for fire service only. This sediment is likely to settle in the sprinkler pipes and either clog them completely, or, if the sprinklers are open, seriously interfere with their discharge. If water is drawn from the bottom for mill purposes the tank may be empty when needed for fire service. Furthermore, the fluctuation in water level is apt to result in shrinkage of the upper ends of the staves of wooden tanks, causing

leakage and hastening corrosion in the steel tank by the repeated wetting and exposure of the sides to the air.

42 Manufacturers in general have awakened to the advisability of following the best practice in the designing and construction of tanks and towers as represented by the foregoing, and are now making structures which are serviceable and safe.

## DISCUSSION

J. W. Ketler[1] (written). Improperly designed supports for tank structures on buildings and the sub-foundations are likely to endanger life and property. As an engineer making a specialty of this work I come in contact with faulty designs, principally in the building itself. The fault does not lie entirely with the structural engineer, but in a great many cases with the architect. I would suggest stringent rules governing the installation of these outfits, especially the thorough examination of the building by a competent engineer or architect familiar with this class of work. I would also suggest, and would coöperate to further the acceptance of one uniform set of specifications governing the engineering manufacture and erection of towers and tanks, both on the ground and on buildings, by all insurance companies, and so far as possible by individuals, corporations or city building departments.

B. A. Freeman[2] (written). Lack of uniformity in the design of tanks and towers at present is well known to those interested in that line of work. Besides the cause suggested, close competition, the lack of ability on the part of most purchasers to discern the merits and demerits of designs submitted helps not a little towards that end. There is little novel or unusual in the construction of these structures. All of the problems connected with them have been solved before in connection with other engineering works. Now that elevated tanks and towers have come into general use for fire protection purposes, it is advisable both to protect the purchaser and the bidder who wishes to erect a good structure, that a specification be forthcoming which shall place all competitors on the same basis. A worthy attempt in that direction has been recently made by the engineering department of the Associated Factory Mutual Fire Insurance Company of Boston,

[1] Chief Engineer, Wondnagel & Co., Chicago, Ill.

[2] Engineer, The Rusling Company, New York.

and intelligent inspections in connection with such a specification should produce very good results.

BRYAN BLACKBURN[1] (written). Too much credit can not be given to Mr. Teague and his associates for the splendid service rendered along the line of tank betterment. The specifications issued by his office, if honestly carried out, will afford a high-class structure that is simple in design, economical in construction and very efficient in service. I have designed a vast number of elevated tanks for fire protection, and I can say without fear of contradiction that the tanks built under the requirements of Mr. Teague's office are beyond criticism.

I differ with Mr. Teague, however, in that it is allowable to single-rivet the vertical seams above the first course. As pointed out in my article on this subject in The Engineering Magazine,[2] the vertical seams should be double riveted, not so much for strength of joint, as to prevent the breaking of the calking edge by breathing tendency of shell due to change in water level; also it is my opinion, based on experience, that the riveting should be closer than is theoretically required for efficiency of joint, in order to insure that the plates be well drawn together under field riveting. The joints in the main column at strut points should be milled to insure full bearing.

The formulae and stresses set forth in the paper for main members are correct and ample, but I would suggest that, while the struts should be of such section as to provide for all live and dead loads, and that the depth of strut should be such that the unit stress due to weight of member should not exceed 4000 lb. per sq. in., still my experience indicates that more often the sizes of these struts are fixed by the requirement that no strut shall exceed 150 radii of gyration in length.

In the using of the Bethlehem H column for main columns, as suggested by Mr. Teague, care should be exercised to see that the column lengths do not exceed 125 radii of gyration. The standard practice in these elevated tanks allows very long column lengths in the latticed channels columns and this has led some inexperienced designers to employ H shapes in too long lengths, with results that are not pleasing, and in several cases that have come under my observation, partial failures have resulted, not from any inherent defect of the H sections but from misuse.

[1] Assistant Engineer, R. & D. Cole Mfg. Co., Newnan, Ga.

[2] Elevated Tanks for Fire-Protective Service, p. 385.

Stress must be laid on the fact that the so-called balcony used on hemispherical-bottom steel tanks is not an ornament but a horizontal girder subjected to large loads due to the horizontal component of the stress in the column.

A. H. Hayes (written). It is generally conceded that in all structures on which the lives of men, or the safety of property from destruction are dependent, too much care can not be taken in design and construction; this is the business of the designing engineer. It is also his duty in this day of efficiency, so to plan his structure that he may use no more material than is necessary to take care of the maximum loading which may occur, with, of course, the proper factor of safety.

In the latter part of the section on towers, the author states that "the total wind pressure on the posts, struts, wind rods, ladders and riser boxing is assumed at 200 lb. per lineal ft. of height of tower." This is taken to mean that for all towers carrying tanks, whether of 10, 20, or 100,000 gal. capacity, the wind pressure must be assumed to be the same. This appears inconsistent, inasmuch as the size and shape of the members will vary with the style of construction, and the loads they are to carry.

The writer has, in several years' experience, designed many towers to carry the smaller tanks on which the wind pressure, taken at 30 lb. per sq. ft. of flat surface, did not reach 150 lb. per vertical ft.; also others of larger capacity, where the wind pressure greatly exceeded 200 lb. per vertical ft. This is a small consideration when designing low towers, but with comparatively high towers where the wind stresses at the base of the tower equal or exceed those caused by the weight of tank, tower and water, the difference amounts in some instances to 15 per cent. Therefore, a strict economy of material is not possible.

I wish to present for consideration another point along this line not brought out in the author's paper, but which is embodied in the Specifications for Gravity Tanks and Towers, recently published by the Associated Factory Mutual Fire Insurance Companies, with which specifications, it appears, the author is largely to be credited; that is the practice of assuming that the wind pressure on the tower is the same when acting in the direction of the diagonal of the tower as when acting in the plane of the bents. This is not correct when the posts are made up, as with many of the smaller towers, of two angles

placed corner to corner forming a star section as illustrated in Fig. 4. Here the wind surface, when the wind is acting in the direction of the diagonal of the tower is $\frac{S}{\sqrt{2}}$ Therefore, the maximum compression stress in the post on the leeward side of the tower, due to wind on the posts, is exactly the same as when the wind is acting in the plane of the bents. The pressure in the struts, wind rods and riser boxing is the same whether the wind acts in one direction or another, and the practice of increasing these stresses by the square

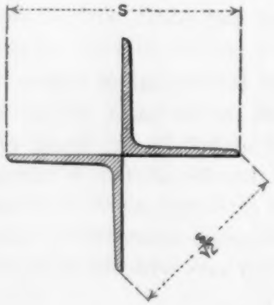



Fig. 17 Surface Exposed to Wind in Post Formed of Two Angles

root of 2 is correct. But in some towers the wind pressure on the posts is the greater part of the total. Take a case where 50 per cent of the total wind pressure is on the posts; then instead of increasing the total wind stresses by the square root of 2, only 50 per cent should be thus increased.

Taking into consideration the fact that an assumption of 200 lb. per vertical foot in many cases raises the total actual stresses in the posts 15 per cent, and that the latter assumption increases them another 10 per cent, it appears that the author has gone even beyond the limits of conservatism, and in order to meet these requirements the manufacturer is compelled to use considerably more material than is needed. This matter should be left as an incentive to each individual engineer to design his structure so as to present the least possible wind surface, and thus produce an economical design.

Albert Blauvelt (written). I take issue with the statement recommending a water level indicator to be placed indoors. Such an

indicator is limited to showing the water level in the tank and may not do that because the operator can not be relied upon to use the test cock alluded to, to prove that the connections are clear. As a matter of field service, it is preferable to use a sight overflow pipe placed vertically inside the tank, with the upper end at the desired full elevation of water and the lower end extending through the bottom of the tank and angled over for easy seeing. By such an indicator, much more is accomplished. The operative condition of the means for filling the tank is proved, the operator can not guess that the tank is full enough; he must fill the tank to get an indication, he must look at the tank structure and thereby is more or less compelled to inspect it, and if the heat has failed and the tank frozen, the top of the sight pipe freezes shut and the tank gives warning of ice by overflowing around the top of the tank instead of at the sight pipe.

G. A. Smith[1] (written). The paper is principally an abstract of the author's specifications covering gravity tanks and towers prepared for the Associated Factory Mutual Fire Insurance Companies of Boston. There are a few points in them with which I do not fully agree, but on the whole consider that the specifications cover the field in very good shape. There are two points, however, which I might mention and on which I would like to have the discussion of different members of the Society:

In the section on Towers for Elevated Tanks, Mr. Teague gives a formula for unit stress in water tower columns. This formula gives a unit stress of 12,000 lb. on the basis of 90 radii. It has always been my contention that a higher unit stress is permissible in the design of water tower columns for the reason that the entire load is quiescent and is entirely dead load. There might be some question raised on this point due to the fact that a certain portion of the column load is caused by wind pressure; however, it is the general practice to permit a higher unit stress for wind loads than for either dead or live loads.

It is a general practice of most engineers in the design of steel structures, such as bridges and buildings, to use a higher unit stress for dead loads than for live loads. For instance, Theodore Cooper specifies for bridges a unit stress in chord compression members of 15,000 lb. for dead load and approximately 7000 lb. per sq. in. for live load, both of these stresses being based on 90 radii. On a similar

[1] Manager, Des Moines Office, Des Moines Bridge & Iron Co., Des Moines, Ia.

basis, I believe that a unit stress of 15,000 lb. per sq. in. for water tower columns is conservative. I would mention, however, that on account of there being so much sentiment against this high unit stress, the company which I represent has adopted a unit stress of 12,000 lb. per sq. in. for columns in their standard water tower specifications.

Another point on which I do not quite agree with Mr. Teague is the expansion joint illustrated in his paper. The illustration is not quite clear as to the kind of metal used in the different parts, but it is his requirement to use cast iron in the pipe sleeve and the collar, and brass or bronze in the follower and adjustment bolts. I contend that this expansion joint will not permit proper operation after it has been used for some time. It is my idea that the cast-iron sleeve will corrode or rust more or less above and below the gasket, and in fact will corrode to some extent on the surface covered by the gasket. After this sleeve has become more or less corroded, I believe that it will not slide freely through the gasket, and in fact there is considerable danger that the riser pipe would probably give in some other point before sliding in the expansion joint as intended. I cannot see why the brass follower gland used by Mr. Teague is any better than a cast-iron gland for the reason that the clearance between the sleeve and the follower is sufficient to prevent any serious adhesion by corrosion.

My idea of an expansion joint which would always work freely, would be to have a brass covered sleeve instead of cast iron so that the surface coming in contact with the gasket would not be affected by corrosion and therefore would slide freely through the gasket.

C. S. Pillsbury[1] (written). Mr. Teague has done a great deal to bring about the general adoption of proper methods in the design and construction of sprinkler tanks, and his specifications are generally recognized by the manufacturers as fairly representing the best present practice in that class of work. In this paper, he outlines a number of suggestions as to proper unit stresses, plate thicknesses, etc., all of which tend to result in a safe and economical structure. The provisions which prohibit bolted connections and reduce the shear on field rivets are especially good.

As Mr. Teague says, it is of great importance to design the details so they will develop the full strength of the main members. To those familiar with railroad bridge shop practice, this statement will seem

[1] Chicago Bridge & Iron Works, Chicago, Ill.

superfluous, but, as a matter of fact, practically every failure of an elevated tank can be traced to an eccentric top post connection, insufficient provision for the horizontal thrust at the top of the posts, poorly made column splices, or some other oversight due to inexperience. To prevent excessive stresses in the tank shell, the number of posts must be proportioned to the diameter and depth of the tank, and provision must be made to resist the torsional moment due to the curvature of the sides. It is also necessary to provide for a number of severe forces other than gravity and wind loads. This last statement is well emphasized by the illustration of a water tower loaded down with ice due to the continuous overflowing of the tank in cold weather. It is only a carefully designed structure that can undergo such treatment without injury.

Following are stress formulae used by the Chicago Bridge & Iron Works which may be of interest in connection with Mr. Teague's paper:

*Tank Sides and Bottom.* The stress in pounds per linear inch in the sides of a cylindrical tank is $2.6 \text{x} H \text{x} D$, where $D$ = the diameter of the tank in feet and $H$ = the head of water in feet. The maximum stress in a hemispherical bottom is $1.3 \text{x} H \text{x} D$ and in an elliptical bottom $2.3 \text{x} H \text{x} D$. The last two formulae are closely approximate, $H$ and $D$ being the same as before, except that in this case $H$ should be taken as the total depth of the tank.

*Posts.* The vertical component of the dead load post stress is equal to the total water and metal load divided by the number of posts. The vertical component of the wind stress equals the following:

3-post tower. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . $\dfrac{M}{0.75\,D}$

4-post tower. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . $\dfrac{M}{1.00\,D}$

6-post tower. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . $\dfrac{M}{1.50\,D}$

8-post tower. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . $\dfrac{M}{2.00\,D}$

where $M$ = moment of wind about panel point at the bottom of the post section considered and $D$ = diagonal of tower at the panel point about which moments are taken.

*Rods.* There is no dead load stress in the rods. The wind stress equals the following:

3-post tower........................ 0.500 ($V$-$V'$) Sec. $A$
4-post tower........................ 0.707 ($V$-$V'$) Sec. $A$
6-post tower........................ 1.000 ($V$-$V'$) Sec. $A$
8-post tower........................ 1.307 ($V$-$V'$) Sec. $A$

where $V'$ = vertical component of post stress in panel above;
$V$ = vertical component of post stress in same panel as the rod,
and $A$ = angle rod makes with the vertical.

*Struts.* Except where the batter of the posts changes there is no dead load stress in the struts. The wind stress is approximately the horizontal component of the rod stress in the panel below.

THE AUTHOR. As stated in the paper and as emphasized by Mr. Ketler, the possibility of danger to life and property from elevated tanks is greatest when they are erected above buildings. I fully agree with Mr. Ketler that buildings should be carefully examined by a competent engineer to ensure that they will safely carry the load to be imposed upon them. His suggestion that a standard specification should be adopted by all parties interested is a timely one, as the recently developed specifications of the Associated Factory Mutual Fire Insurance Companies can be used as a basis, such changes or additions being made as seem necessary so that the standard will thoroughly cover all conditions. It would be desirable to have this Society assume the leadership in the work.

Mr. Freeman also advocates uniformity in the design and construction of tanks and towers, and recommends a standard specification following along the lines of those of the Factory Mutual Companies.

In reply to Mr. Blackburn's suggestion that the vertical seams above the first course be double riveted, I have not considered it necessary to advise this, since experience with the single-riveted joints has thus far been quite satisfactory. Certainly if much trouble from leakage at these joints should be experienced, the double riveting would be advisable.

Mr. Hayes has brought up a point which should receive more careful treatment, i.e., the wind load on the tower members and pipe fittings. I have followed current practice in assuming the total wind pressure on these members as 200 lb. per linear foot height of tower and have not, as yet, attempted to vary the pressure to suit various

heights and types of towers, although I believe that this is advisable, especially in the case of towers to support tanks for other uses, such as water works' supplies. The 200 lb. pressure gives reasonably accurate loadings for sprinkler tank towers, as the height of these is fairly constant within limits. A closer approximation of the loading, even for these towers, would, however, be appreciated by the designing engineer.

The further point which Mr. Hayes mentions, that the actual wind pressure varies according as to whether the wind blows at right angles to plane of the bent or diagonally is, of course, correct, especially for angle-iron towers. As a further refinement in determining the wind load on the structure, it would be well to give this matter consideration.

Mr. Blauvelt's comments regarding the use of the overflow to determine if the tank is full, represent the best practice, and I would say that the use of a mercury gage as a telltale is intended merely as a constant indication of the water level, in place of the unreliable float and target telltale and other devices.

In reply to Mr. Smith, 12,000 lb. per sq. in. is considered a proper maximum allowable unit stress in compression for tank towers by most manufacturers and engineers. This stress is none too conservative, in view of the very special type of the structure where failure of one post member results in immediate and complete collapse. Furthermore, the towers are exposed to the weather, and as they are usually the property of comparatively small companies unfamiliar with the proper care of steel structures, they will not be inspected and kept painted as will steel railroad bridges and other structures, and so the steel members should have some additional thickness, in order that they may remain serviceable even after some considerable corrosion has taken place.

I agree with Mr. Smith that a brass or brass covered pipe at the expansion joint is preferable to the iron pipe and believe that this would remove the last possibility of binding through corrosion in the present design. This improvement may be secured in time, but the expansion joint shown is so much better than what has been used that it seemed wise not to add further to its cost at this time, since there undoubtedly would have been serious objections raised by the manufacturers for commercial reasons. The advantage of the brass follower is, of course, its non-corrodibility, which will permit its being moved to tighten the packing even if the iron pipe is corroded, as

mentioned by Mr. Smith. The oil lubricant with which the packing is saturated will naturally prevent any appreciable corrosion of the pipe where it bears on it and so permit of movement of the pipe without injury to the packing.

The suggestions made by Mr. Pillsbury are important and the data which he has given regarding stress formulae will undoubtedly be useful.

No. 1424

# FIRE PUMPS

By Ezra E. Clark,[1] Boston, Mass.

Non-Member

Were we to review historically the development of fire pumps, we would find a long list of efforts, dotted frequently with failures, and marked here and there with a partial success. No sooner does man meet fairly well the fire pump requirements of an age than he finds he must make a still better apparatus; for in the march of human progress, the needs of fire protection always keep just a little ahead of the provision and the vigilance of man. We could find much of interest in this buried history. We would find how the masterpiece of some old-time mechanic had been the cherished marvel of a community and had been forced by a rival into a state of obsolescence. Even the fire pump of today will sometime become an obsolete device. Nevertheless, it represents the present state of the art. It is the pump on which we depend to stiffen the pressure of a weak public water supply. It is the pump from which we expect prompt and reliable service in the extinguishing of fires. It is the business man's concern to know something about these pumps, their possibilities, their record, which one of the various types should be selected for a given situation, and how to install them so as to secure their best service. Are not these vital questions for the man who expects to spend a limited appropriation for fire protection? He may ask, "Why do I need a fire pump when I have a tank on my mill tower and a 6-in. connection to the public main?"

2 An answer to his question may be found in the history of the Paterson, N. J., fire, eleven years ago, where a group of six mills with their own fire pump protection, checked completely the conflagration sweeping towards them. Each mill had its own fire pump and a stored water supply, the supplies being supplemented in some cases by a connection to the public mains. So intelligently were these moderate fire equipments handled that when the fire's defeat was made

[1]Engineer and Inspector, Factory Mutual Fire Ins. Cos.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

certain, there was still water to spare in the pump cisterns, and the mills had sustained no material damage. In another instance, quite recent, where there was no pump but simply a tank and a moderate public water supply, the fire ran so rapidly through the building that 50 sprinkler heads opened, and several fire streams were quickly brought into service. These combined drafts on the water supply

Fig. 1 View of Fire Streams from Centrifugal Pumps

so reduced the pressure that neither sprinklers nor fire streams were at all effective, and the building was destroyed with a loss of $8000, the value of several fire pumps. There are very many instances that could be cited where the prompt starting of a fire pump has assured ample pressure, a good distribution of water from the sprinklers or several effective fire streams, and a comparatively small loss.

3 The argument is being advanced today that because 90 per cent of our fires are being extinguished with perhaps a few sprinkler heads, rarely requiring the service of a fire pump, insurance interests are demanding supplies of water far more generous than is reasonable, and that much smaller supplies and smaller pipe connections would amply suffice to give reasonable protection. This logic is attractive to the man who pays for the equipment. But fires have a way of proceeding along illogical lines and it is the wise man who holds himself

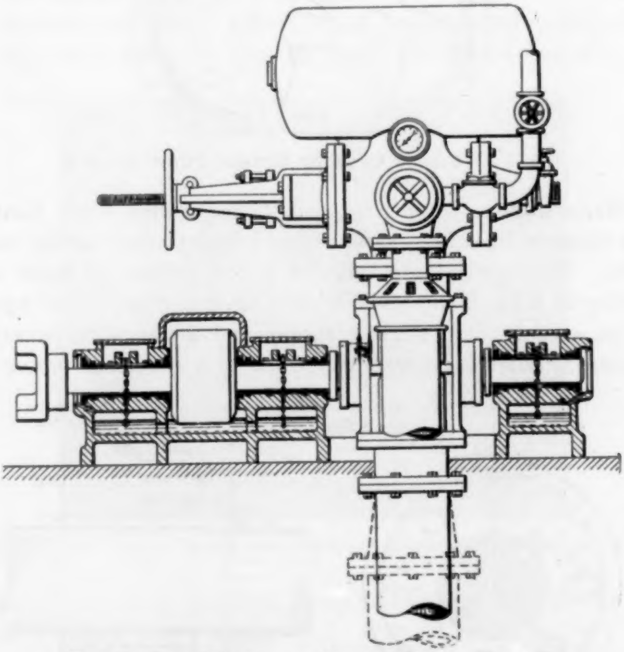


FIG. 2 PART SECTION OF ROTARY PUMP, TYPE B

prepared for the illogical and the unexpected condition, even for the worst conceivable condition, the doubtful 10 per cent.

### DEFINITION OF TERM

4 Broadly speaking, any device by which water can be thrown on a fire is a fire pump. In its simplest form, it is a bucket of water and a man. In the earliest days the bucket was supplemented by huge syringes in the form of a bladder or bellows, the gut of an ox serving as a length of hose. Later they were made from brass cylinders and

were called "squirts." From these crude appliances were developed the hand and steam fire engine around which cluster the memories of many exciting contests during the last century. These devices, however, are all portable in character, and valuable time is often lost in bringing them to the scene of action.

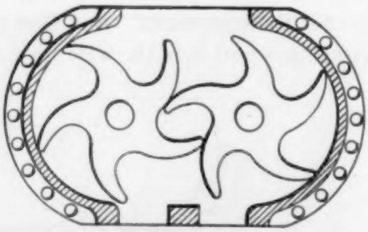


FIG. 3 WORKING CAMS OF ROTARY PUMP, TYPE B

5 Manufacturing properties, particularly, were often located at such a distance from fire engine stations that prompt service was impossible. The incalculable value of a few gallons of water at the beginning of a fire became self-evident, so that pumps fixed upon the premises, connected to a water supply and a source of power, were introduced. Later these were connected to a system of piping about

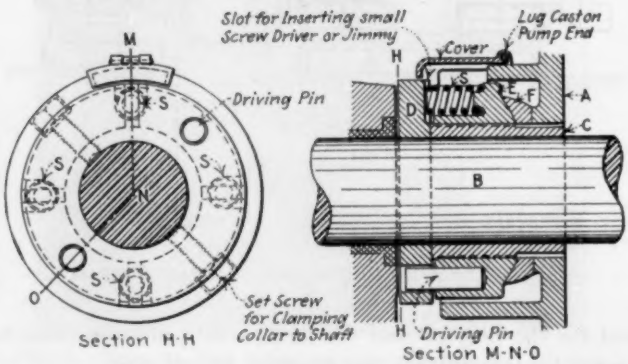



FIG. 4 WATER JOINT USED ON ROTARY PUMP SHAFTS, TYPE B, AS SUBSTITUTE FOR STUFFING BOX

the yard and in the mill, and the pump then became available to supply sprinkler heads, or furnish fire streams from hydrants.

**6** In this paper, the term fire pump is to be used in its restricted sense, i. e., a pump that is installed in a fixed position for fire purposes. In this sense there are three or four distinct types, generally acknowl-

edged: rotary, duplex, centrifugal, and power pumps, any one of which, of course, could serve as a fire engine by mounting it on wheels and providing a source of power.

TYPE OF PUMP

7 The type of pump to be selected should be determined by the character of power available for the purpose. Naturally the most reliable source of power at any situation would be chosen, and then a pump selected that can best be driven with that power. This should be the general rule. Every type of pump, however, has its own limitations and these must be kept in mind when studying the whole

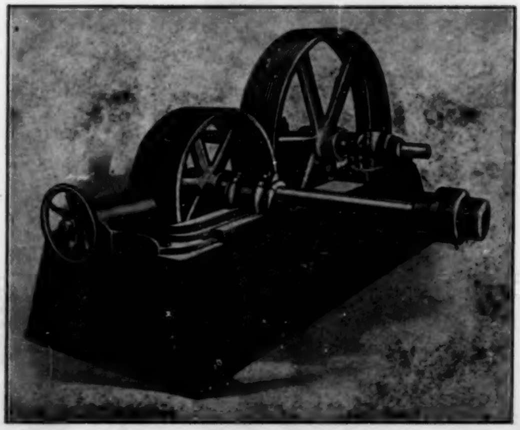


Fig. 5 Typical Press Frame for Friction Gears driving Rotary Pumps

problem. The natural selection would be a pump that can run at or near the speed of the prime mover. One instance has been noted where a centrifugal pump was connected to a waterwheel through a pair of spur gears having a ratio of 5 to 1, and another case where an electric motor was connected to a rotary pump through spur gearing having a ratio of 1 to 5. In each case, the pumps took their suction supply under a head. Both cases may be dismissed as examples of poor engineering, for the requirements of fire service would have been much better served if the two pumps were to exchange places and the reduction gears omitted, thus securing the simplest, cheapest and most reliable arrangement.

8 The earlier mills were operated almost exclusively by waterwheels and their usual speed made it comparatively easy to connect

with rotary pumps. Thus, a great many pumps of this type found their way into the wheel pits of mills, although they were too often, unfortunately, poorly located. Power plunger pumps were used in a few cases, but the rotary came to be the prevailing pump, owing to its simplicity, cheapness and adaptability to the prevailing power. Where waterwheels of ample power and moderate speed are available, the rotary type of pump becomes the logical choice, for the two can often be direct-connected, or some simple form of transmission, such as

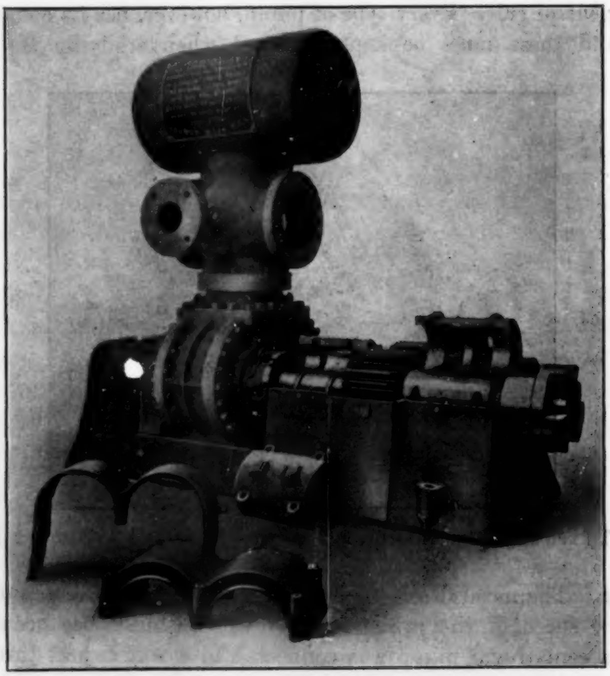


FIG. 6 ROTARY PUMP, TYPE B, PARTLY DISMANTLED

friction gears, or a silent chain having a low speed ratio can be employed. The impulse form of waterwheel with its high speed would find a better selection in the centrifugal pump, requiring no intermediate gearing. In weighing the claims of the several pumps for a situation, three things should be kept in mind: the character of the power available, the limitations of the pump and the fact that the simplest drive is the best.

## ROTARY PUMPS

9 Among the earlier types of pumps installed in mills for fire purposes was the rotary. There are so many indifferent rotary pumps in mills and factories that the mere mention of a rotary raises doubts as to its value for fire purposes. Still, it should be remembered that even with the imperfect construction of the old time cast-iron rotary, they have given some very effective service at critical times. However, I invite your attention not to these obsolete pumps, but to that product of the rotary pump builder that meets the present insurance specifi-

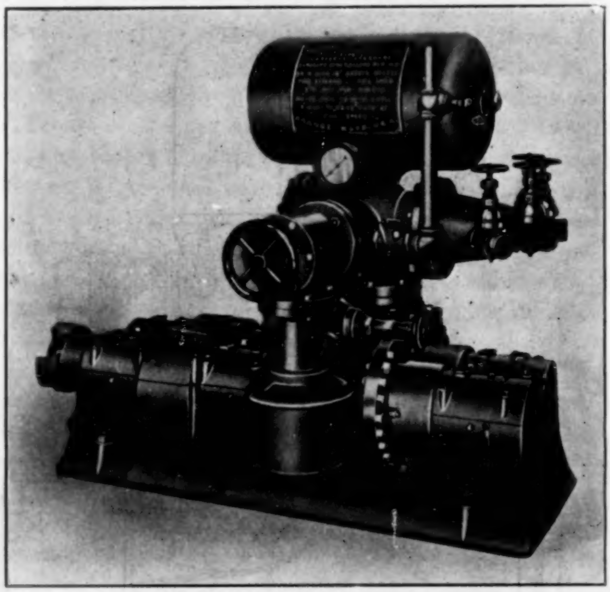


Fig. 7 Rotary Pump, Type B, with Standard Fittings

cations and notably to that which is designated as Type *B* (Figs. 2 and 3).

10 Several features of this design were proposed and developed by the author in 1904. They were not immediately adopted, as some experimental work was necessary, and new patterns were required. These new features were chiefly as follows: In place of the usual two sets of overhung cast-iron gears, one extra-heavy pair of cut steel gears is substituted, preferably forged solid with their shafts; these gears are supported on either side by generous bearings, and the two shafts are larger and stiffer; all bearings are provided with liberal

oil reservoirs and ring or chain oilers; in place of the usual stuffing boxes, a special form of metal water joint has been adopted, which is giving excellent service, thereby avoiding the use of perishable packing. The usual fire pump features are also provided; viz., the casing and working parts exposed to corrosion are of bronze; there are more liberal water passages; the working cams are of a new design to insure reasonably smooth running; and the usual air chamber, hose connections, an approved spring relief valve and starting valve complete the arrangement. The press frame carrying the friction gear is now fitted with a heavy spring that maintains the gears in contact,

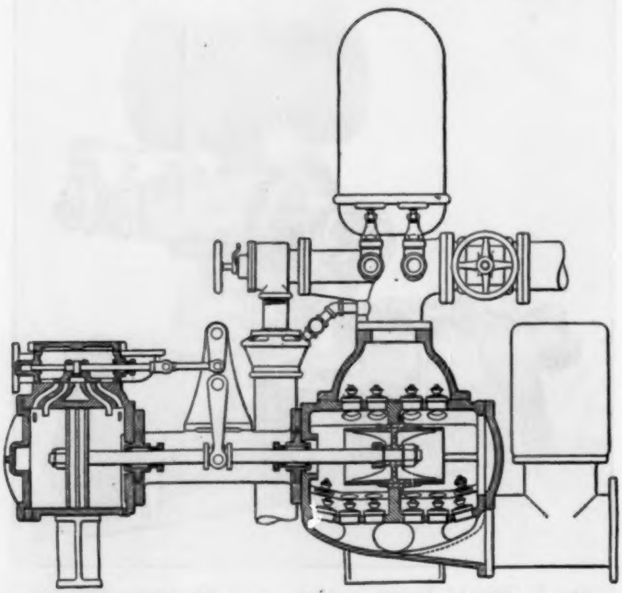


FIG. 8 SECTIONAL VIEW OF UNDERWRITER STEAM FIRE PUMP

and compensates for a possible lack of truth in their running (Fig. 5).

11 These pumps are not intended for daily service as the wear between cams and casing would soon render them inefficient. But for the brief periods of fire service, considerations of wear are of less importance than certainty of action. Full particulars as to their construction can best be gathered by a study of published specifications. The effort has been to secure a rugged, reliable pump capable of doing hard work, resisting corrosion and withstanding abuse. These pumps are obtainable today from at least two manufacturers, and cost

somewhat less than a duplex pump. The limited number that are already in the field are so far giving excellent satisfaction.

### DUPLEX PUMPS

12 Where steam is the most reliable source of power, the duplex "Underwriter" pump is usually the choice, because of its record of proved value. It is also much the cheapest of any steam-driven fire pump of acceptable reliability, selling for approximately ten cents a pound. In spite of its numerous parts, liable to derangement, this pump has been so thoroughly boiled down, so to speak, that with present day construction it is proving to be one of the most reliable of fire protection appliances. The steam-turbine driven centrifugal pump is, of course, available for those who are willing to pay the price, but it costs approximately twice as much. Its main advantage over the duplex is its simplicity of working parts. Its disadvantages lie in the fact that a pressure of 100 to 125 lb. of steam is required for good results, whereas, with a duplex pump, 50 lb. of steam is ample for ordinary fire pressures. The outfit is also subject to the same limitations as to suction lift as all centrifugal pumps. When the steam turbine centrifugal has become standardized and its price made more nearly competitive with that of the duplex, no doubt it will receive a much wider introduction.

13 As steam began to supplement and in cases to supplant water power in mills, steam-driven pumps were installed to help out or replace the rotary. At first the single cylinder pump was used, but the advent of the duplex pump marked the beginning of a better machine for fire purposes. The duplex type of fire pump is so well known that it is hardly necessary to describe it, full information as to construction being available in published specifications. When the development of the Underwriter type of duplex pump was undertaken by Mr. John R. Freeman in the early nineties, he found the pump as made for the trade lacking in several features needed for fire protection. The frequent failures in the field mainly showed that improvement was needed. But pump builders did not fully sense the importance of some of the features that were demanded, and more or less opposition was encountered.

14 The old trade duplex pumps were found with steam and water passages restricted, but these have been enlarged in the Underwriter type, and a higher speed thus made possible. The trade pumps were iron fitted and became rusted through neglect, so that they could not be started; the Underwriter pump is brass fitted wherever

corrosion is liable to seize the working parts. In the older pumps, cast-bronze valve stems were used that worked loose or broke off; in the present Underwriter standard, these stems are of rolled bronze and made heavier, and the guard is so secured in place by a nut lock devised by the author that the parts cannot work loose (see Fig. 9). In the old trade pump it was common practice to provide means of adjustment for the valve motion, and this adjustment sometimes worked loose, or was tampered with, putting the pump in a lame condition; in the Underwriter design, all adjustment has been ruled out and a simpler and more reliable mechanism is now used. Special fire service fittings are attached, such as priming valves, starting valve, relief valve, hose valves, oil pump and lubricator, all these parts having been subjected to examination and test before adoption. (See Figs. 8 and 10.)

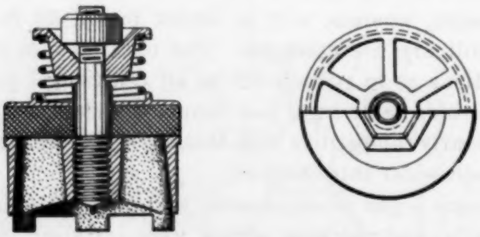


Fig. 9 Detail of Valve Stem and Special Nut Lock

15 To incorporate these changes has taken time and it has been more or less an up-hill job. Experience, however, has shown the value of these improvements and has indicated from time to time still further improvements which have been adopted, so that today we have a fire pump that is able to start at a moment's notice, run at full speed if needed, and when supplied with steam, water and oil, will run continuously and deliver its rated capacity, or more, against a fire pressure of 75, 100, 150, or even 200 lb. in special cases.

16 A fire pump to meet these conditions must be rust-proofed. It must be built of material chosen for strength, toughness and durability. It must be so free from complications as to permit of operation by men of moderate ability. It must be able to withstand safely a large measure of abuse, and be, so far as possible, "foolproof." This is the sort of steam pump that is being furnished for fire protection today. It is not claimed that the pump is built like the wonderful

"one hoss shay," equally strong in every part, but rather from specifications that make the possibility of failure extremely remote.

### ELECTRICALLY DRIVEN PUMPS

17 Where the electric motor is the most reliable source of power, the centrifugal pump naturally becomes the choice. The speed of a

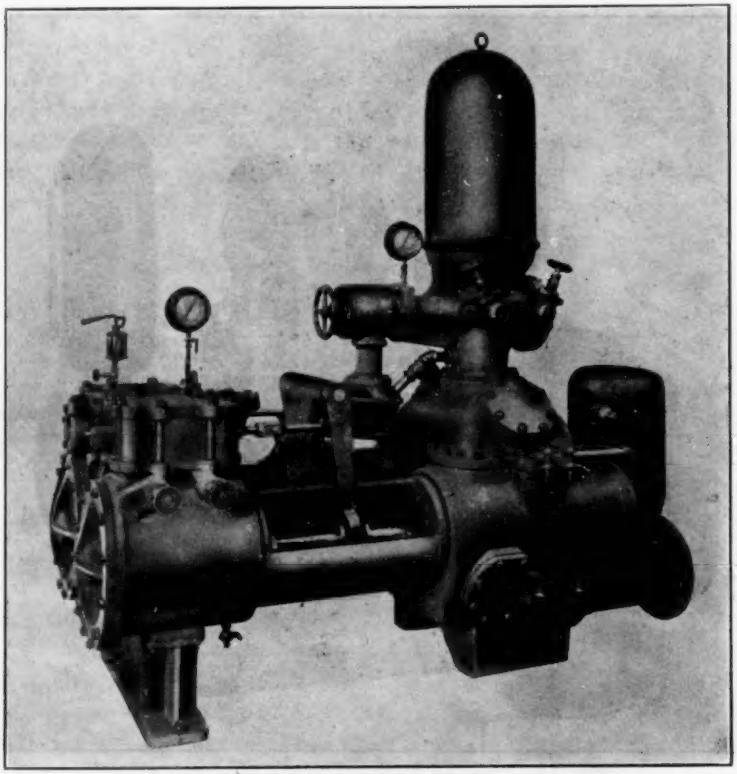


FIG. 10 GENERAL VIEW OF UNDERWRITER STEAM FIRE PUMP

centrifugal pump can readily be made to conform to the speed of the standard motor and all intermediate gearing avoided. The direct-connected motor and pump thus form the simplest possible arrangement. It has, however, certain limitations which will be noted later, and there may exist conditions which some form of power pump electrically driven would meet with better satisfaction.

18 Comparatively few power pumps are in use as fire pumps, owing to their highest cost. Their best field is where a daily service pump of good efficiency is needed, that can in the emergency be used for fire purposes. The triplex type of power pump is much to be preferred to the single or duplex forms, owing to its steadier discharge pressure, and as it is usually built in the vertical form the floor space required is the minimum. A strong point in favor of the triplex power

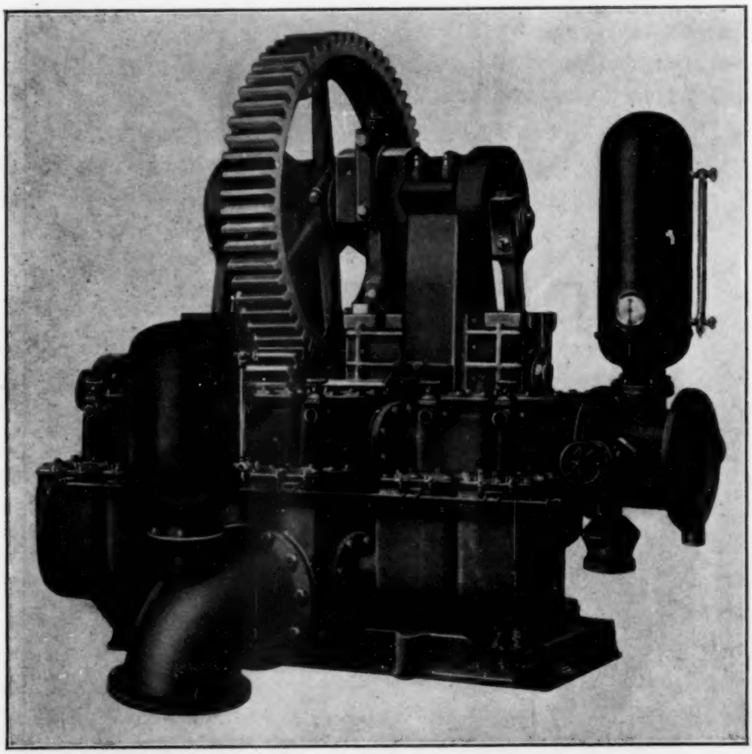


FIG. 11 VIEW OF TRIPLEX POWER FIRE PUMP

pump as compared with a centrifugal is its higher efficiency, running as high as 80 to 85 per cent for outside packed plungers as against the usual 65 to 70 per cent for the centrifugal. That type of triplex pump employing one centrally located driving gear is much to be preferred to those having two overhung gears. Experience has shown that the central gear drive will better ensure an even distribution of power, and thus avoid undesirable stresses on gear teeth and possible

breakage. A notable example of the central gear triplex power pump may be found in the high-pressure fire service station in Philadelphia, where seven such pumps are operated by gas engines, and have given excellent satisfaction with a minimum of repairs.

19 The centrifugal pump is comparatively a newcomer in the fire pump field. It is distinguished from all others in being a *velocity* pump, the pressure varying directly as the square of the peripheral velocity of the impeller. The electric motor is no doubt responsible for its coming so rapidly into favor. The constant speed motor direct-connected to the pump makes the simplest and cheapest arrangement and affords a fairly wide range of capacities and pressures. The variable speed motor or steam turbine, costing a little more, would place the pump more nearly on a par with the steam duplex as to range of pressures, and, the author believes, would make the preferable

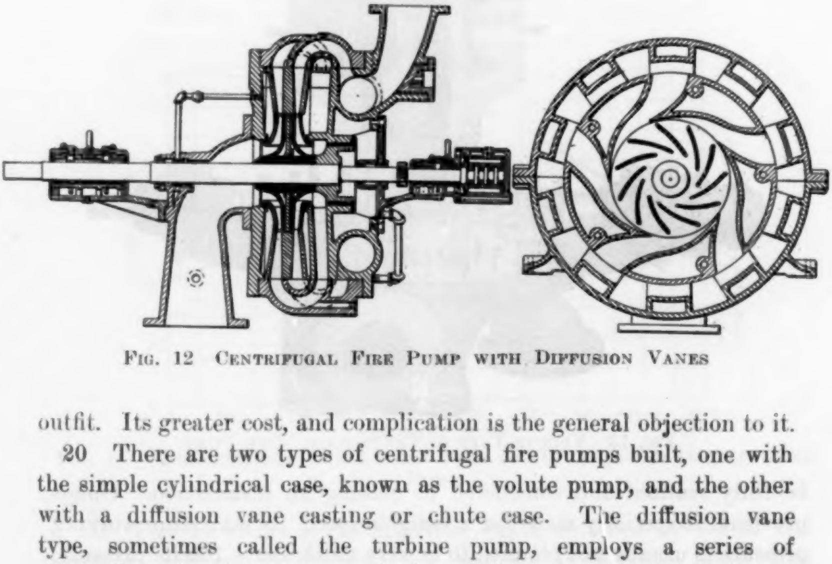


Fig. 12 Centrifugal Fire Pump with Diffusion Vanes

outfit. Its greater cost, and complication is the general objection to it.

20 There are two types of centrifugal fire pumps built, one with the simple cylindrical case, known as the volute pump, and the other with a diffusion vane casting or chute case. The diffusion vane type, sometimes called the turbine pump, employs a series of gradually enlarging passages that permit a more gradual change of the velocity of the water into pressure than does the volute, thus resulting in some slight gain in efficiency. This, however, is not an essential part of an acceptable fire pump, and as the volute type of pump (Fig. 13) is much simpler in construction, and yields efficiencies that are satisfactory (60 to 70 per cent), this type of centrifugal fire pump is being generally adopted. This type also lends itself better to

the horizontal division of casing, which is preferred to the circumferential division, as it permits quicker and better access to the interior for overhauling or cleaning.

21 The simplicity of the centrifugal, having but one moving part, appeals to every engineer. There are no valves to choke up, no plunger to wear out, no valve motion or gears to break, and no dangerous pressures possible even with all outlets closed, unless specially provided with a variable-speed prime mover. The discharge is steady, smooth, devoid of shocks, and more nearly approximates that from a gravity supply than any other pump. But it has its limitations, which should

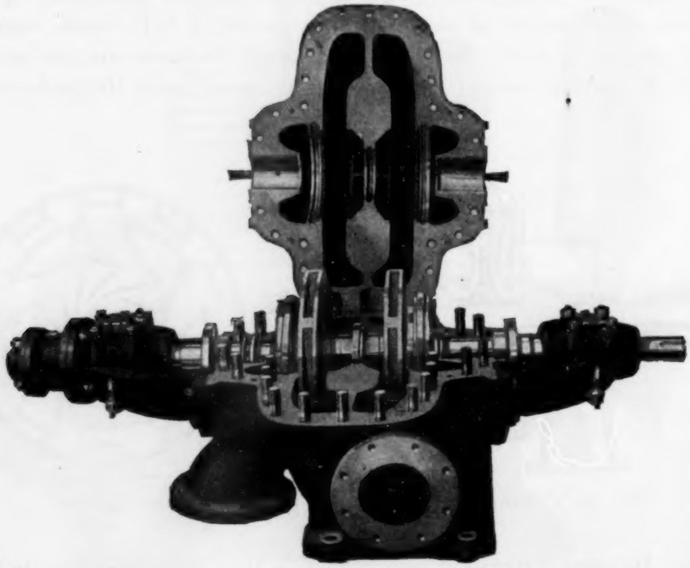


FIG. 13 VOLUTE TYPE OF CENTRIFUGAL FIRE PUMP

be fully realized and considered in making an installation. Unless provision is specially made for a variable speed, its maximum working pressure is usually not planned to be very much above 100-lb. pressure, running for the smaller flows in some cases up to 130 or 140 lb. Consequently the water pressure needs of a situation should be carefully studied before deciding on this form of pumping outfit, and if higher pressures are needed, either a special form of impeller should be substituted or a variable speed provided.

22 A centrifugal has no power to exhaust the air from a suction pipe, as has the rotary, duplex, or other displacement pump. For this

reason, the suction supply should come to the pump under enough head to flood the pump casing, thus insuring its being primed. If a lift is unavoidable, then it becomes necessary to provide a suitable foot valve and a generous supply of priming water, enough to fill completely the suction pipe and pump casing. This feature, of course, limits the centrifugal to situations where there are easy suction supply conditions.

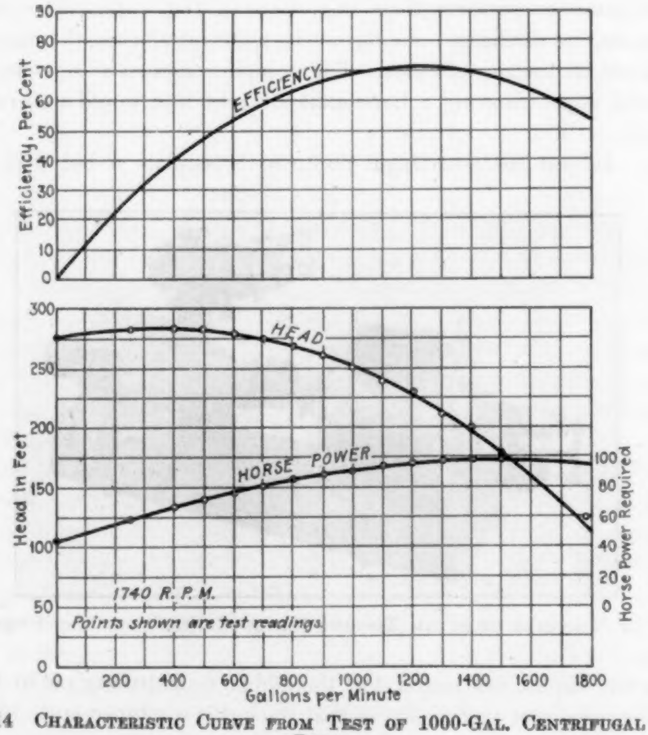



FIG. 14 CHARACTERISTIC CURVE FROM TEST OF 1000-GAL. CENTRIFUGAL FIRE PUMP

23 Fig. 14 shows a characteristic curve obtained by actual test from a 1000-gal. centrifugal fire pump. It will be noted that at its rated capacity of 1000 gal., its discharge pressure runs about 108 lb., and at half capacity, or 500 gal., it runs as high as 123 lb., while at 1500 gal. the pressure drops only to 78 lb. Thus with a constant speed motor it is possible to get a good range in discharge capacities, 50 per cent above and 50 per cent below the rating, and all at pressures that would be serviceable for fighting fire.

24 The curve shown is approximately what is desired and obtained in the average two-stage centrifugal. For some situations it may be desirable to secure a different form of characteristic, and this curve can be made steeper or flatter within limits.

25 Where two pumps are to operate together these curves should be similar and the characteristic curves should be known and posted in view of the operator, in order that he may intelligently operate each pump to perform their fair share of the work. The shut-off pressure (no discharge) should not be materially below the maximum pressure, at the top of curve. For pumping against a constant head, a curve approximating a horizontal straight line would naturally be chosen.

26 In cost the centrifugal pump with bedplate is but little more

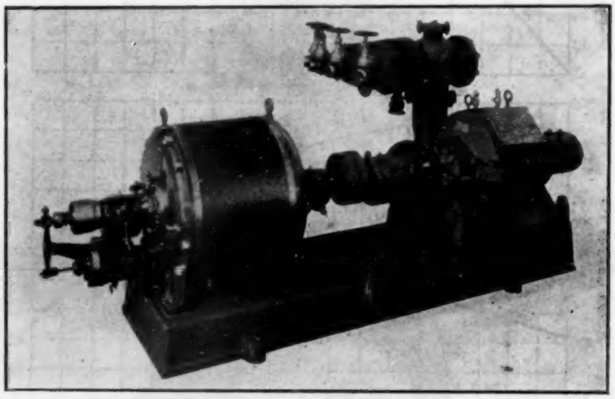


FIG. 15 EXAMPLE OF STEAM TURBINE DRIVEN CENTRIFUGAL FIRE PUMP UNIT

than the duplex per pound, but the addition of driving motor brings up the price very materially, so that the outfit combined costs 15 to 16 cents per lb., and weighs about the same as a duplex pump. For a given type, the speed of a motor determines its size and influences the total price of outfit. It may roughly be stated that the centrifugal pump and motor costs 50 per cent more than the duplex. There is a wide difference, however, in the weights of the same size pump among different manufacturers.

### INSTALLATION

27 In a brief paper like this, the details of a pump's installation cannot be properly reviewed, and yet the success or failure of a fire

pump depends very much on its being properly erected and connected up. Briefly, it may be said that a fire pump must be so located as to be safe at all times from the breakage of its pipe connections due to the falling of floors and machinery, and safeguarded from any influence such as smoke, fire or flood that might drive away the operator. It should be accessible at all times for operating and overhauling, and not blockaded by miscellaneous storage. It must permit of practicable pipe connections, the suction pipe receiving first consideration, and the steam and discharge pipes so run as to be safe from damage and always be in commission. The exhaust should go direct to atmosphere, and not be tied up with other pipes.

28 The suction supply should preferably be practically inexhaustible, such as a river or lake, and a suitable intake, properly screened, provided. Where a stored supply only is available, it should be large enough to supply the pump for two hours, more or less, depending on conditions. No cast-iron rules can well be established as the insurance engineer having jurisdiction is expected to weigh conditions, and secure a reliable pump service without expensive or complicated refinements. Full details as to pump installation have been covered in the several publications and specifications which from time to time have been developed in the Factory Mutual Inspection Department.

29 It will perhaps be of interest, and not altogether without value, to look a bit into the future and try to discern what is to be the development of the fire pump. It is not at all unlikely that the steam turbine centrifugal will gradually displace the duplex pump, as its design becomes simplified and standardized, and its cost lowered. Where steam is not available, we shall find the electric-driven centrifugal, or the gasolene-driven rotary, and as fast as its development permits, the gas-turbine-driven centrifugal will possibly win recognition and receive adoption. To whatever extent insurance interests adopt these new appliances, it will be done with an eye single to their proved value, as regards simple and rugged construction and reliability of performance.

## DISCUSSION

Alfred B. Carhart. Upon the practical side of the operation of fire pumps, I think the author of this paper has called attention, in one of the final paragraphs on Installation, to a very important consideration, which was not mentioned in the presentation of the paper.

It is such a simple matter, that it seems surprising that engineers in laying out new plants, will connect the exhaust of steam-operated fire pumps with other exhaust lines, and then supply stop valves by which these pipes can be cut off from the exhaust line to prevent back-pressure and condensation, not realizing that when the fire pump is needed in a hurry, it will be impossible to operate it effectively, because there will be no free outlet for the exhaust steam. Such conditions have been discovered, much to the chagrin of those in charge of otherwise admirably constructed plants.

Albert Blauvelt (written). Referring to the section of Mr. Clark's paper on electrically driven pumps, it is my observation in the field that the electric motor centrifugal pump is as yet considerably short of working development as compared to the Underwriters' steam fire pump. The steam pump is a self-contained affair including the more essential fittings, and ordinarily is handled successfully when needed. The electrically driven pump thus far appears always to require considerably more skill and sense of time on the part of the operator. The motors are excellent and so are the pumps, but neither pump maker nor motor maker appears to know each other, nor do either appear prepared to deliver a complete pumping set, fully fitted and adapted to unfamiliar and incompetent handling in a degree comparable to the steam pump. Meantime, mismoves, or delay or total failure of starting, mark the attempt of watchmen or other men not trained to the electric pump. I take no pride in the various electric pumps installed in our practice thus far, and have less esteem for similar jobs set up under other auspices. Doubtless as the trade increases some shop will put out a complete and self-contained electric pump, with fittings interlocked or timed against mismoves.

The Author. From the remarks of Mr. Blauvelt, it appears that he finds that men not trained to the electric pump make mismoves, or totally fail to start the pump. This is also true of men not trained to steam or rotary pumps.

It should be remembered that the steam fire pump has been growing up with us for 40 or 50 years, and there is a larger number of average men familiar with such apparatus than there is familiar with the electrically driven centrifugal pump. The latter has been with us but a few years, and for the average man it is still somewhat of a novelty that he but vaguely comprehends.

A remedy for this situation does not appear to be getting the pump maker to know better the motor maker. It is rather for the pump operator to know better the apparatus under his charge. It has not been found necessary for rotary pump builders to furnish the water-wheel driving the pump, nor has it been necessary for steam pump makers to go into the steam boiler business.

The troubles arising from centrifugal outfits are mainly due to errors in installation, or a failure to realize the limitations as well as the possibilities of this type of pump.

A properly installed centrifugal and motor is one of the simplest of fire pumps, and easy to start. Regarding interlocking devices to prevent mismoves, these are but complications that add to the confusion of the operator. A simple device under intelligent control is much to be preferred to automatic control involving added complications. It is not necessary for a man to be an electrician in order to operate a motor driven centrifugal, any more than it is necessary for a steam pump operator to be an expert in hydraulics, pneumatics, and the use of steam. But he does need to know something about these matters in either case, and especially to know the use of the fittings and devices that he sees before him. There is no positively certain way to prevent an incompetent man from making the fool move.

There will no doubt be improvements made from time to time based on experience tending to standardize these outfits, which will simplify their operation; but the lack of confidence in present apparatus which is expressed by the gentleman is not shared by the author.

## No. 1425
# NECROLOGY

Edward Miner Adams was born in Pittsburgh, Pa., July 19, 1868, and went to Crestline, Ohio, when two years of age. He had only a common school education, being obliged to go to work in the Pennsylvania Railroad shops when thirteen years old. Here he learned boiler making and the coppersmith's trade. Later he went to Bucyrus, Ohio, where he was employed by the Thompson Steam Shovel Company as machinist. In 1892 he moved to Akron, Ohio, and worked as machinist with the Akron Machine Company. After three years he accepted an offer from the American Cereal Company, later the Quaker Oats Company, and in 1898 attained the position of machine shop foreman. In the same year he was made chief engineer in addition to his other duties, a position he retained until the time of his death on April 17, 1913.

Mr. Miner was president of the Ohio Society of Mechanical, Electrical and Steam Engineers, having been chosen to serve a second term; also a member of the Akron Chamber of Commerce.

### SIR WILLIAM ARROL

Two factors will always be found in any engineering structure of magnitude. The first is the scientific or design factor supplied by knowledge of physical law. The second is the craftsmanship, or the experience and skill evidenced in embodying the design in material form. The first requirement is for technical knowledge and experience. The second calls for knowledge of men, experience, credit and administrative capacity. In the United States the economic trend has been in the direction of meeting the need of the hour by means of standard constructions. The design and the engineering skill are both supplied by the constructor from within his organization and the constructor as contractor supplies the finished work to the user. This makes for economy in many directions. The British system has been to have the design and the engineering skill and responsibility furnished by one party who will usually be called a consulting engineer, and the craftsmanship and execution of the work confided to some one entirely different. The consulting engineer sees

only that his design is properly carried out. The skill and economies of the process of construction and erection are confided to the contractor. The latter makes the money if there is any profit to be made.

Sir William Arrol and the Dalmarnock Works near Glasgow, Scotland, furnish a noteworthy example of the best application of the British principle. Sir William owed his advancement to his splendid ability and courage in the attack of big problems, and his success financially to his attacking these problems in a large way, and applying machinery on a large scale to effecting economies in construction. His engineering office was not engaged on problems of outside design. It was directed to the making of the details which were required to produce economically and effectively the material and the erection of the designs of others.

He was born in 1839. His father was a Scotch cotton spinner and rose to become manager of the great Coats Thread Works of Paisley. William entered the works under his father's management as a boy in the woodworking shop, but at the age of fourteen he evidenced his preference for metal working and he entered the blacksmith shop of Mr. Thomas Reid. An apprenticeship lasted four years in those days and during a period of six years thereafter he traveled as journeyman in various parts of Scotland and England. At twenty-four he became foreman of the bridge and boiler departments of the works of Laidlaw & Sons, remaining with them for five years. In 1868, or when he was twenty-nine years of age, he invested his savings of 85 pounds sterling in a little plant in the outskirts of Glasgow for machinery repairing, boilermaking and general blacksmithing. More than one-half his capital went into the engine and boiler alone. Business must have come to the young energy and enthusiasm of the little shop, for extensions became necessary and it was in 1871 that the Dalmarnock Works were begun with about thirty men. The ability and capacity which have come with opportunity developed that modest beginning into the plant of today, covering some twenty acres and employing when in full operation about 5000 men.

The foundation of his fame as a bridge builder and constructor was early laid by a long multiple-span bridge for the Caledonian Railway across the Clyde. He introduced in these bridges what was at that time the entirely novel method of erection on the ground, constructing the span without false work and rolling each girder out over the piers until it could be dropped into position. This method

attracted the attention of all the engineers and work immediately began to flow in. The viaduct over the Clyde at Glasgow gave opportunity for his design and application of multiple drilling in the field and the hydraulic riveter. This was in 1875.

It will be recalled that in 1879 England experienced one of its great engineering disasters in the wreckage of the bridge over the Tay. The bridge had to be re-erected and Mr. W. H. Barlow was entrusted with the work and made Mr. Arrol his contractor. The rebuilding of the bridge took five years and was completed in 1887 with 74 spans and a total length of 10,700 feet. It has withstood all attacks from the tempestuous gales from the North Sea to which it is exposed. The success of this undertaking led Sir John Fowler and Mr. Benjamin Baker in coöperation with Mr. Arrol to undertake the great cantilever to cross the Firth of Forth. Tunnels and suspension designs were carefully investigated, but while a cantilever principle was decided upon as the most feasible, the dimensions were far in excess of anything that had been attempted before. The two main spans are 1710 feet long with a headway below the center span of 150 feet above the water. The towers are a little above 360 feet high. The scale of the work to be done called again for special plant and the value of the investment on the ground has been estimated at over a million dollars. It was at this time that Mr. Arrol produced an oil fired furnace for heating rivets and greatly improved the hydraulic riveting machines. It was at the Forth Bridge that Mr. Arrol invented what has been called the hydraulic spade to work in the resistant boulder clay into which the caissons had to be forced. Queen Victoria conferred the honor of knighthood on the successful contractor on the opening of the bridge in 1890 and in that same year he received the honorary degree of LL.D. from the University of Glasgow. The Forth Bridge was still building when the contract came for the steel work of the Tower Bridge of London with its 200 feet clear opening and its lifting bascules of 1200 tons each. In addition Sir William's firm did a great deal of viaduct and swing bridge work for the Manchester Ship Canal, for bridges over the Nile at Cairo, for a bridge with the widest span in England over the Wear at Sunderland, and elsewhere both in England and in the colonies. The organization of the Dalmarnock Works was made into a limited liability company in 1895 and the building of steel factory and railway structures was made a part of the business. Sir William was a great believer in abundance of light as a factor in successive

and rapid production, and favored the modern steel glass roof construction.

Fame came to him in other directions and he sat in Parliament from 1895-1906. He was not a talker but was much relied upon by his colleagues for his sound advice in industrial legislation in committees. He was a Fellow of the Royal Society of Edinburgh and other scientific and philanthropical organizations. The Institution of Engineers and Shipbuilders in Scotland honored him with its Presidency in 1895 and in that same year The American Society of Mechanical Engineers elected him an Honorary Member. The letter which started that movement in the Society was signed by Mr. Andrew Carnegie, who had long admired the man and his methods.

His personal character was one of rare charm, sympathetic, tactful, warm-hearted and generous. His capacity for attention to detail was renowned and his energy phenomenal. He died after several weeks' illness from an attack of influenza on February 20, 1913.

F. R. H.

## JAMES RICHARD BELL

James R. Bell, who died at his home in Hazeldene, Ightham, Kent, England, on July 14, 1913, was born on August 21, 1841, in Wick, Scotland, in the house of his grandfather, James Brenner, a famous engineer. Mr. Bell was educated at the Aberdeen Gymnasium, and served his apprenticeship with the Gardiners in London. His early work was on various surveys in England and Wales and on the construction of the Cairo Ramelah Railway.

In 1868 he went to India, being connected first with the irrigation department in the Madras Presidency, and in April, 1870, was transferred to the state railways directly controlled by the Government of India, where he remained until his retirement in 1896.

Mr. Bell's most important work in India was in connection with the building of railways and bridges, and he devised many novel means for the training and improvement of diluvial rivers, on which he was a recognized authority. Among his achievements are the Empress Bridge over the Sutlej River, the Aneroati Railway, the railway to Hyderabad, Deccan, the Jumna Bridge, near Mutra, an important bridge over the Sutlej at Ferozepore, and a bridge over the Chenab River at Sher Shah, near Multan. Mr. Bell also investigated the bridging of the Indus at Sukkur, where a steam train ferry had been provided for the Indus Valley State Railway, and over

which Mr. F. E. Robertson subsequently constructed the cantilever structure now standing.

In 1891 Mr. Bell became consulting engineer to the Government of India for state railways, receiving a permanent appointment to this position in 1894, which he held until his retirement in 1896.

On his retirement Mr. Bell was still in excellent health and he practised as a consulting engineer. He is said to have been the original of Findlayson in Kipling's "Bridgebuilders," and was regarded by his contemporaries in India as an engineer of extraordinary attainment, though his services were never more than formally acknowledged and never adequately honored by his government. Apart from the successful and very rapid construction of several fine railway bridges and the evolution of the only proper and scientific method of controlling the flow of itinerant rivers when bridged, he deserved the most substantial recognition of his work on the Ruk-Sibi Railway during the Afghan War, not only because through the exercise of marvellous organizing and directing capacity he performed almost a miracle in railway construction, but because he provided possibly the only means of ending a critical situation in Afghanistan.

### ADOLPHUS BONZANO

Adolphus Bonzano, pioneer bridge builder and inventor of the Bonzano rail joint and other railroad appliances, was born at Ehingen, Germany, December 5, 1830. He received a classical and engineering education both at Ehingen and at Stuttgart, and in 1850 came to America to perfect himself in the study of English. In 1851 he went to Springfield, Mass., where for the following four years he served as apprentice, machinist and draftsman for the American Machine Works. Until 1860 he was engaged as superintendent of construction of the Detroit Dry Dock Iron Works, which was later transformed into the Detroit Bridge & Iron Works, one of the earliest bridge building plants in this country. In 1868 he moved to Phoenixville, Pa., where with Thomas Curtis Clarke and others he formed the firm of Kellogg, Clarke & Company, bridge builders, acting himself as chief engineer. In 1884 this firm was dissolved and was succeeded by the Phoenix Bridge Company, of which Mr. Bonzano was made vice-president and chief engineer. In 1893 he resigned this position and with Mr. Clarke opened a consulting engineering office in New York. After his partner's death in 1898, Mr. Bonzano retired from active business to devote himself to

the invention of railroad and other appliances. The Pecos viaduct on the Southern Pacific Railroad in Texas, the Kinzua viaduct on the Erie Railroad, and the Chesapeake & Ohio bridge at Cincinnati are among the more notable examples of his work. He died May 5, 1913.

### EDWARD L. BRONSON

Edward L. Bronson, who died at his home in Waterbury, Conn., on February 18, 1913, was born at Wolcott, Conn., on May 18, 1860. His education was received in the public schools and in the Lewis Academy, Southington, Conn. Upon leaving school he apprenticed himself to the Hendey Machine Company in Torrington.

In 1886 he entered the employ of the Waterbury Farrel Foundry & Machine Company at Waterbury, as machinist, and later in the same year resigned to take up similar work with the E. J. Manville Machine Company. Three years later he became connected with J. E. Draper & Company, North Attleboro, as a tool maker, but soon returned to Waterbury to become foreman of the machine department of Blake & Johnson, machinists and manufacturers of piano hardware, etc. In 1896 he again entered the employ of the Manville Company, now as foreman of special machinery and tool making, and remained with them until he became master mechanic of the Shoe Hardware Company, with which he was connected until shortly before his death. His was no small part in the development of the factory to its present importance.

Mr. Bronson was especially interested in automatic machinery and with A. C. Campbell invented an improvement in dress hook machines, which resulted in increasing the speed of production from 70 to 220 hooks per minute.

### WALTER S. BROWN

Walter S. Brown, who was drowned while canoeing on the Lehigh Canal, July 13, 1913, was born in St. Louis, Mo., on December 30, 1871, and received his technical education in Washington University. From 1892 to 1902 he was connected with the St. Louis waterworks, first as draftsman, and then as assistant mechanical engineer in the construction and testing department. During part of this time he served as first lieutenant of the volunteer engineering regiment in the Spanish-American War, returning from Cuba after the war in command of his company.

In 1902 he removed to Bethlehem, Pa., to become engineer in the power department of the Bethlehem Steel Works, and held this

position at the time of his death. Mr. Brown was a member of several masonic and benevolent orders.

### WILLIAM GEORGE CHAMBERS

William George Chambers, chief engineer of the National Cash Register Company, Dayton, Ohio, died in New York on May 1, 1913.

Mr. Chambers was born at Belleville, Ontario, on April 2, 1874, and on completion of his education in the public schools, served his apprenticeship with the Bagley & Sewall Company, Watertown, N. Y. In 1900 he entered the employ of the National Cash Register Company as a model-maker in one of the inventions departments. The following year he won promotion and became foreman of one of the assembling departments. In 1905 he was sent to Canada as superintendent of the company's factory at Toronto, returning the following year as foreman of another assembling department. Two years later he was made supervisor of the entire assembling division and in 1910 became assistant general superintendent. When the important position of chief engineer of the company was created in 1912, Mr. Chambers was named to fill the place, becoming responsible for the ordering of all changes and improvements in registers, the quality of the material used, and all the inspection, tool designing and toolmaking of the firm.

### GEORGE W. CLANCY

George W. Clancy, president of the Globe Chemical Company of Boston, died at his home in Albany on March 10, 1913. Mr. Clancy was born in Albany on February 22, 1881, and after completing his education in the public schools entered the shops of Skinner & Arnold of that city, where he learned the machinist's trade. He then entered the employ of the New York Central at West Albany and was transferred to the Boston & Albany Railroad in 1904, becoming inspector of shops. In 1908 he left to accept a similar position with the New York, New Haven & Hartford Railroad. While with this company he received an offer to become manager of the railway sales department of the Adams & Elting Company, Chicago, and was with this concern at the time of his death.

### TRENMOR COFFIN

Trenmor Coffin was born in Carson City, Nevada, November 4, 1886, and received his early education in the Carson City schools. He was awarded an appointment to the Naval Academy at Annapolis,

but after a year and a half left there to enter Purdue University, from which he was graduated in the class of 1910. He returned to the University the following year as instructor in mechanical engineering, resigning in January 1912 to become instructor in manual training at the Indianapolis High School. In 1913 he transferred to the University of Texas as instructor in mechanical engineering and manual training. While here his health failed and he died on his way to his mother's home in Reno, Nevada, on February 16, 1913.

### EDWIN S. CRAMP

Edwin S. Cramp, until recently vice-president and general manager of the William Cramp & Sons Shipbuilding Company of Philadelphia, Pa., was born in Philadelphia, March 1853. He was the son of Charles H. Cramp and the grandson of William Cramp, founder of the great shipyard that has been known as "the cradle of the navy." He obtained his education in the Philadelphia public schools, graduating from the Central High School in 1871. In the fall of that same year he was apprenticed to the engineering department of the Cramp company and served his four years there, becoming an expert machinist. He then left the machine shop for the drafting department where he spent seven years, familiarizing himself with all details of naval designs. Following this he was placed in charge of the erecting department, erecting all marine engines used by the company, and was subsequently made superintendent of the plant. From this position he rose to general manager of the entire shipyard and in October 1901, on the death of Henry W. Cramp, he became vice-president of the company.

He retired from business a few years ago and moved to New York, where he died on June 20, 1913. His work side by side with his father for 35 years was effective in making the Cramp company foremost in the production of our "new navy"; his technical and practical training was the cause of exceeding specifications in many trial trips of completed vessels which were run by himself personally and netted the company large sums as bonuses. Mr. Cramp was vice-president of the Society from 1896 to 1898 and was identified with the Pennsylvania Society, American Academy of Social Science, the Geographical Society, Naval Architects and Marine Engineers, and the Engineers Club of Philadelphia.

### FRED H. DANIELS

Fred H. Daniels was born at Hanover, N. H., June 16, 1853. Immediately after graduation from Worcester Polytechnic Institute

he entered the employ of the Washburn & Moen Manufacturing Company as draftsman and came under the direct supervision of Charles H. Morgan, general superintendent of the company. Soon afterwards he was transferred to the machine shop and in this department of the works obtained an experience which proved of infinite value in his later career as an engineer and manufacturer of wire rods and wire.

About the year 1874 his company thought it imperative that a laboratory in charge of a competent chemist be added to their works, and decided to send Mr. Daniels to study under Thomas M. Drown of Lafayette College, Easton, Pa. Here he remained 18 months, and then was sent, in company with Mr. Morgan, to study the state of the wire industry in England, France, Germany, Norway, Sweden and Belgium. He returned to Worcester in April 1876 and again took charge of the drafting room and laboratories, where he inaugurated new and advanced practice based upon his investigations. In 1378 he accompanied Mr. Morgan abroad a second time to inspect a continuous mill with horizontal rolls in Germany. Practically all continuous mills heretofore had alternating horizontal and vertical rolls, a combination possessing great disadvantages. Upon their return to Worcester they were granted patents on the horizontal rod mill as well as on the reels, and a mill was built which proved to be very successful. The tonnage was doubled and rods two sizes smaller than had ever been rolled were produced. The invention revolutionized the art of rod rolling throughout the world. Both the reels and the mills are now almost in universal use, reducing the cost of rolling $2.50 per ton.

Shortly afterwards Mr. Daniels was promoted to the position of superintendent of buildings, retaining charge of the drafting room and laboratories. In 1887 upon the resignation of Mr. Morgan from the company, P. W. Moen was made general superintendent and Mr. Daniels assistant general superintendent of all the Washburn & Moen plants. About this time the business had grown to such proportions that it was decided to open a second plant at Waukegan, Ill. Mr. Daniels designed and superintended the construction of the buildings having an output of 800 tons per day of all kinds of wire rods and wire. After acting as assistant general superintendent for a year, he was promoted to general superintendent, and in 1895 was sent to San Francisco, where in conjunction with Frank L. Brown, Pacific coast sales agent, he established the Hallide works,

known as the California Wire Works. In 1898 the Washburn & Moen Manufacturing Company was taken over by the American Steel & Wire Company and Mr. Daniels became engineer of their 30 plants. He was given the responsibility of putting a number of run-down properties into first class condition, and the business was then sold to the United States Steel Corporation. Mr. Daniels was appointed chairman of the board of engineers, in charge of 143 plants.

During 1902 and 1903 Mr. Daniels designed and constructed at San Francisco the Pacific coast works of the American Steel & Wire Company. In 1907 he was delegated to go abroad by his company to inform himself fully regarding rolling mills for producing hot rolled flats. Simultaneously the United States Steel Corporation authorized him to negotiate for the purchase of a shop right for its plants in the patented device of O. E. Theisen of Munich for purifying blast-furnace gases.

In 1907 and 1908 he designed and constructed for his company the Cuyahoga works at Cleveland, and in 1910 and 1911 their Birmingham works.

Mr. Daniels was a members of the American Institute of Mining Engineers, the American Society for Testing Materials, the Iron and Steel Institute of Great Britain, the American Iron and Steel Institute, and was a Life Member and Past Vice-President of the Society. He died on August 31, 1913.

### CARL GUSTAF PATRIK DE LAVAL

On February 3 of this year there passed away an engineer of world-wide fame, Dr. Carl Gustaf Patrik de Laval, whom we are proud to have numbered among our honorary members. He was in his sixty-eighth year, in the full possession of those faculties, the conscientious use of which in the service of mankind had placed him in the front rank of his profession.

Dr. de Laval was born at Blosenberg in the Province of Dalecarlia, Sweden, on May 8, 1845, of a family of soldiers whose French ancestors rode to fame with Gustavus Adolphus. The inherited qualities of courage, alertness, endurance, and close observation, he carried into that nobler warfare with stubborn materials and elemental forces in which the wit of man conquers not by destruction but by making their energies his willing instruments.

In his eighteenth year he entered the technical department of the University of Upsala, and was graduated with distinction three years

later, carrying with him, indeed, the highest honors in every department. His first position was with the Stora Kopparberg Company of Bergslad as a draftsman, but considerations of health forced him to give up this confining occupation for one with more opportunities for outdoor exercise. He accordingly decided to return to Upsala for further study, and in 1872 received the degree of Doctor of Philosophy. In the same year he re-entered the employ of the copper company and made for them investigations in the manufacture of sulphuric acid. Upon completion of this task Dr. de Laval resigned his position for a venture of his own, a glass factory at Falun, which, however, unfortunately failed, leaving him heavily in debt.

He then entered the employ of the Klosterverken Iron Works as engineer, and it was during his connection with this company and while carrying on his daily duties in the works that he developed from a crude German device the centrifugal cream separator which has made his name a household word on the dairy farms of the world. So absorbed did he finally become in its perfection that he gave up his position in order to devote his entire time to it. When it finally worked to his complete satisfaction he endeavored to secure in Stockholm the financial assistance needed to place it on the market, but it was only by extreme persistence that he was able to obtain even a small loan. Its manufacture proved an immediate success and it was not long before the Separator Company, Limited, was on a secure financial basis.

The cream separator called, however, for some means to drive it at the high speed required, and Dr. de Laval attempted to apply to it the ancient principle of Hero of Alexandria, driving a reaction wheel by two steam jets normal to the direction of rotation. This in its turn brought the inventor face to face with troubles with transmission and shafts rotating at high speeds, which he attempted to overcome by using a special bearing, lubricated by the oil acted on by centrifugal force, together with a high-speed worm-drive transmission. For many years cream separators were made with direct steam drive, but their economic efficiency proved to be poor. At the end of the eighties de Laval was ready with the first steam turbine, embodying practically all the features which have since become familiar to every engineer, such as conversion of the energy of high-pressure steam into velocity by means of the de Laval nozzle, the use of a wheel running at a very high speed, and the flexible shaft running at a speed far exceeding the critical speed. The new prime

mover, running at what was then considered a frightful speed of some 24,000 r.p.m., which had to be reduced to one-tenth of it for driving the separators, did not prove convenient for this purpose and became the basis of a separate industry.

Dr. de Laval was not the type of man to rest content with one success and to devote his time to the management of his existing concerns, even though that clearly appeared to be the easiest way to wealth. In looking for new things to which to turn his inventive genius, de Laval started the manufacture of milking machines, and when this enterprise proved unsuccessful, devoted his time and fortune to discovering a new process for treating low-grade Swedish zinc ores. This venture, which gave some interesting scientific results, proved so disastrous to him financially, that in 1908, at a time when the Separator Company in which de Laval was originally one of the largest shareholders, was more prosperous than ever before, his affairs were in such a state that the company voted him a pension of 12,000 krones.

Large rewards came to him in wealth and honor, but, instead of hoarding his gains, he saw in them only the means and the incentive for further work for the benefit and advancement of mankind. In the ethical, no less than in the intellectual sense, he deserves the title of a great engineer. Dr. de Laval was a member of the Swedish House of Lords and during his lifetime received many foreign orders and distinctions.

E. D. M.

### CHARLES SIMEON DENISON

Charles Simeon Denison was born at Gambier, Ohio, July 12, 1849, and was educated at the high school at Lockport, N. Y. He spent one year at Norwich University, Northfield, Vt., where he acquired a military training. In 1871 he was graduated from the University of Vermont with the degree of C.E.; in 1874 he received the degree of M.S.; and in 1907 the honorary degree of Sc.D. was conferred upon him by his alma mater.

He was assistant engineer in construction of the Milwaukee and Northern Railroad from 1871 to 1872, and since then had been instructor at the University of Michigan, rising through various positions to head of the mechanical drawing department, which he held at the time of his death on July 30, 1913.

Mr. Denison was United States astronomer and surveyor in locating the boundary line between Washington and Idaho territories in

1873 and 1874. The accuracy of his work upon this survey, accomplished largely in the winter under severe conditions and great hardships, was fully established by a recent expedition sent out by the United States Government for placing permanent markers on the boundary. He was a member of the Society for the Promotion of Engineering Education, the Michigan Engineering Society, Detroit Engineering Society, and an honorary member of Tau Beta Pi, and was the author of numerous published essays and lectures on scientific subjects. He had been an extensive traveler in America and Europe, largely in the interest of his work at the University of Michigan.

### RUDOLPH DIESEL

The recent death of Dr. Rudolph Diesel is a severe loss to the engineering profession at large, as well as to the many personal friends of the distinguished engineer. Dr. Diesel, who had been spending a week-end at Ghent with M. Carels of Carels Frères, sailed from Antwerp in company with M. Carels and M. Luckmann, chief engineer of the firm, on the evening of September 29, with the intention of attending the annual meeting of the Consolidated Diesel Engine Manufacturers, Ltd., in London. Upon arrival in Harwich next morning, it was discovered that Dr. Diesel was missing, having apparently fallen overboard during the voyage. Some days later his body was recovered, thus confirming the fears of his family and friends.

Dr. Diesel's career has been one of interest from the first. In the early eighties of the last century, as a student in the Technical High School of Munich, while listening to a lecture on thermodynamics by the famous Professor Linde, he jotted down in his notebook these words: "See if it would not be possible to realize practically the isothermal." It was only a carelessly scribbled line, but in it was the germ of a great idea, which in the mind of a man peculiarly fitted to "see" was destined to develop a dozen years later, into what may be justly considered as one of the most important achievements of mechanical engineering of the last century, the Diesel engine.

Rudolph Diesel was born in 1858, the son of a German family residing in Paris. In 1870 changed political conditions forced his father to move to London and to send the boy to Augsburg, Germany, the very place where, twenty-three years later, the first Diesel engine was placed on the testing stand, to be greeted at first with incredulity and then with the unstinted applause of the engineering world. The

boy's youth was spent, in the methodical German fashion, in years of high school training and thorough drilling in engineering fundamentals under the competent supervision of men like Schroeter and Linde, interrupted from time to time by long visits to London where his father was still making his home and where he acquired that mastery of the English language which was such a pleasure to his many friends in this country.

From the beginning Diesel was kept in close touch with the problems of thermodynamics, first as assistant to Linde in his work at the Munich Technical High School, and later as his technical representative in connection with the manufacture of the Linde refrigerating machinery, then at its beginning. These studies resulted in Diesel's taking out a patent for a new engine in the early nineties, and in his publishing a book under the title, "The Theory and Construction of a Rational Heat Engine." The rational engine was based on a principle so simple that very few consented to believe in its truth, viz., that pure air could be compressed to so high a point that the fuel injected into it would ignite and burn. Happily for Diesel, however, there were a few men like his former teacher Professor Schroeter, of Munich, and Professor Hartmann, of Charlottenburg, who expressed their belief that the engine would work on the new cycle, and their opinion was powerful enough to induce the largest two machine construction companies in Germany, the Krupp works and the Augsburg-Nuremberg Company, to unite in an effort to test out the invention.

Two years of trial followed. The first engine blew up as soon as fuel was injected, nearly killing Diesel himself. Alterations were made, an air supply pump for the injection of fuel added, but the engine would not run and was always a source of danger. "I myself," said Diesel in his talk before the Society in 1912, "would never have had the patience and the courage to continue the work after the disappointments of the first two years of experimenting, had I not been supported by an unalterable belief in the correctness of my mathematical deductions."

Finally, after four years of laborious experimenting, the correctness of mathematical deductions,—the true beacon of all engineering work,—was vindicated in the form of an engine working on an entirely new principle, with a thermodynamic economy practically not yet exceeded. From 1897 the name Diesel became attached to a new prime mover which has since been built in thousands of units.

The vindication of the principle of the Diesel engine led to a strenuous effort to promote its introduction into all fields where it could be used. The development of the horizontal engine, the marine Diesel engine, the high-speed engine for driving dynamos, and finally the Diesel engine locomotive with direct drive and electric transmission are matters of history, and it is to be noted with profound regret that the inventor's returns were by no means commensurate with the services he rendered to humanity.

The members of the Society who visited Dr. Diesel at his delightful home in Munich during the summer of 1913 were enabled to renew the acquaintanceship formed during Dr. Diesel's tour of America in 1912, when he spoke before the Society of his own work. It is seldom that any one has so endeared himself to the members generally as had Dr. Diesel, and his death is a personal loss to the membership of the Society.

### WILLIAM M. DOUGLASS

William M. Douglass, treasurer of the Globe Real Estate Company of Allentown, Pa., died in that city on March 23, 1913. Mr. Douglass, who was one of the early members of the Society, began his professional career in 1878 in the steel mill of the Gautier Steel Company, Johnstown, Pa., where he had partial charge of erecting engines and trains. In the following year he was given the management of the mill and during the subsequent years of his connection with it built one train and rebuilt two others, mainly from his own plans. In 1883 he began the plans for the Hartman Steel Company at Beaver Falls, erecting all the roll trains, engines, boilers, etc., partly from his own designs. In September of the same year he was made superintendent of the entire plant.

In 1884 Mr. Douglass removed to Allentown, to become general superintendent of the Iowa Barb Wire Company, continuing in this capacity until 1894 when he became general superintendent of the Consolidated Steel & Wire Company. In 1899 he became superintendent of the American Steel & Wire Company of the city, advancing in 1904 to the position of assistant to the general superintendent. He retired from the rolling mill field in 1906, and in 1911 entered the business in which he was engaged at the time of his death.

### VICTOR DWELSHAUVERS-DERY

On March 15, Victor Auguste Ernest Dwelshauvers-Dery, professor emeritus of the University of Liège, died in Belgium, bringing

to a close a long life of service in the field of engineering research and investigation. Professor Dwelshauvers-Dery was born at Dinant on April 25, 1836, and receiving his early education at home, entered at the age of seventeen a small college in Brussels in which he gave special attention to the study of higher and applied mathematics. In 1861 he was graduated from Liège University with the diploma of mechanical engineer and in the same year accepted from his alma mater the position of lecturer in mechanics. The university's recognition of his talents was soon justified by his rise to the chair of professor of mechanics, a position which he held for many years.

The important thing to which Professor Dwelshauvers-Dery immediately directed his attention was the establishment of an engineering laboratory where theory might be compared with practice. He established in the university courses of lectures covering in detail the theory and construction of the steam engine and the economics of steam raising, and his object in securing a laboratory was the investigation of certain discrepancies between steam engine theory and practice. He began his efforts toward this end as early as 1870, but it was not for ten years that they were even partially successful, a grant being then allowed him for the purchase of a steam engine, which, however, he had to run at his own expense outside of the university premises. In 1893 the laboratory was finally established according to Professor Dwelshauvers-Dery's original plan, and he immediately began there his experiments with steam consumption and steam jacketing, gathering about him in this work many brilliant students. The effect of high compression on steam economy was one of the earliest matters to be investigated, for which Professor Dwelshauvers-Dery designed an experimental steam engine where he was able to vary at will the steam distribution, the governing and also the degree of compression. From these experiments he obtained his theories on steam compression which are widely known. His views on the phenomena due to steam jacketing under most varying conditions, one of his favorite subjects, were presented at a meeting of the Institution of Mechanical Engineers and are published in their proceedings for 1905.

Professor Dwelshauvers-Dery was elected to Honorary Membership in the Society in 1886, and Professor Thurston, who had come into contact with him through his work on internal condensation in the steam cylinder, was his active proposer. A paper by Professor

Dwelshauvers-Dery on a Contribution to the Theory of the Steam Engine, was presented at the Chicago meeting of 1893, which many foreign authors were invited to discuss.

In 1900 Professor Dwelshauvers-Dery was invited to become rector of the university and was obliged in undertaking these heavy duties to interrupt his work in the laboratory. Three years later he retired from public life, but his advice and collaboration were still given freely wherever asked and he contributed extensively to the Belgium and French technical press.

Professor Dwelshauvers-Dery was honored in 1888 by the award by the Institution of Civil Engineers of the Watt Medal and Telford Prize for his paper on the Steam Engine Governor, and in 1889 he shared with Donkin a prize for investigations on the thermal action of walls of a steam engine cylinder. That his teachings were widespread in their effect is evidenced by the fact that the establishment of the laboratory in University College, London, was due to the work which he had done at Liège. He was a corresponding member of the British Association, the Institut de France, the Société Industrielle de Mulhouse, and the Société d'Encouragement pour l'Industrie Nationale, a commander of the Leopold Order, and a knight of the Legion of Honor.

### W. H. FLETCHER

W. H. Fletcher, vice-president of the W. & A. Fletcher Company, Hoboken, N. J., was born in New York on May 18, 1857, and after an education in the public schools entered the shops of Fletcher, Harrison & Company, of which his father was president.

Besides his connection with the Fletcher Company, he was president of the Consolidated Iron Works. He had been president of the Engineers Club of New York, was secretary of the Robert Fulton Memorial Association, and vice-president of Webb's Academy and Home for Shipbuilders. He was a member of the New York Yacht, Governors Island, New York Railroad, Royal Arcanum and Lotus Clubs. His death occurred on April 2, 1913.

### ROBERT FORSYTH

Robert Forsyth, one of the most widely known marine engineers in the United States, died in San Francisco, December 18, 1912. Mr. Forsyth was born in Lanarkshire, Scotland, on August 18, 1848, and came to San Francisco as chief engineer of the British steamer

Prince Alfred in 1870, and left to join the staff of the Pacific Mail Steamship Company. He was assistant superintending engineer in 1876, resigning this position to become chief engineer of the Risdon Iron Works. In 1886 he became chief engineer of the Union Iron Works, and during his twenty years of service in this capacity and as president of the company more than twenty-three vessels for the United States Navy were constructed, including the famous Oregon and Olympia and a large fleet of merchant vessels.

Mr. Forsyth retired in 1906 to engage in consulting practice. His death removes another of the few remaining who have been identified with the creation of the shipbuilding industry on the Pacific Coast.

### HORATIO A. FOSTER

Horatio A. Foster was born at Bustleton, Philadelphia, Pa., January 12, 1858. His engineering training began in the fall of 1884 with the Daft Electric Company, Greenville, N. J.; the next year he went to Baltimore to electrify a short branch of the Baltimore Union Passenger Railway Company. In 1886 he entered the shops of the Thomson-Houston Electric Company at Lynn, Mass., and in September 1888, was appointed superintendent of the East River Electric Light Company, New York, remaining with that company till July 1891. He was then appointed an expert for the United States Census office to compile data on the electrical industry of New York State. In May 1893 he accepted a position in the editorial department of Electric Power, and later in the same year became associated with George Forbes, electrical engineer of the Niagara Falls Power Company, and had charge of his New York office for about a year and a half. In 1895 Mr. Foster joined the staff of the Cataract Construction Company of Niagara Falls as testing engineer. After several years in general consulting work he became interested in the valuation of public utilities, studying traffic conditions and other matters pertaining to public service, being engaged in this work with J. G. White & Company at the time of his death, April 27, 1913.

Mr. Foster was the author of the Electrical Engineers Pocket-Book, which bears his name, also of a book on the Valuation of Public Utilities, and he had frequently contributed to the technical press. He was a member of the American Institute of Electrical Engineers, the Engineers Club of New York, and the Philadelphia Arts Club.

### JOSEPH HARLAN FREEMAN

Joseph Harlan Freeman, consulting engineer and mechanical expert, died at his home in Brooklyn, N. Y., on January 27, 1913.

Mr. Freeman was born near Farmington, Mich., in 1868, and his boyhood was spent in Grand Rapids, Mich. Financial necessities compelled him to leave school at an early age to enter upon wage earning. He chose the mechanical side of the printing art as his vocation and served until he became proficient in that line in the plant of the Michigan Tradesman, at Grand Rapids.

Realizing the handicap of his lack of schooling he resolved to obtain a better education, particularly along technical lines, and at the age of nineteen years he entered the engineering department of the Michigan Agricultural College, graduating with the degree of B.S. in Mechanics in 1890. Upon completing his college training he became a designer and foreman with the Buss Machine Works of Grand Rapids, where he designed several important new machines during the ensuing year.

From 1891 to 1898 he served as an examiner in the Patent Office in Washington, and during that time, by studying evenings, he completed the full course required for and received the degree of LL. M. from Georgetown University.

In 1898 he removed to New York City and joined the staff of an able firm of patent attorneys. Here, as well as while in the Patent Office at Washington, he specialized in patents pertaining to printing machinery, particularly the mechanism and processes of multi-color printing. After more than four years of highly successful work in the employ of that firm he established an independent office as a consulting engineer and patent expert, which he continued to conduct with marked success up to the time of his death.

### ARTHUR J. FRITH

Arthur J. Frith was born at Philadelphia, Pa., February 23, 1852, and died at Chicago, Ill., November 10, 1913, of acute dilation of the heart. Mr. Frith was educated at Georgetown College, Georgetown, D. C., and at the Rensselaer Polvtechnic Institute, where he was graduated in 1873 with the degree of civil engineer.

As a young man Mr. Frith worked in steel and rolling mills and was sent to England for special information in regard to steel manufacture. He taught at Lehigh University; was assistant chief en-

gineer in connection with the Mississippi River Commission on Government work; designer for the Newark Machine Tool Works, the C. W. Hunt Company, and engineer on coal-handling plants with the Trenton Iron Company; assistant chief engineer and designer with the Diesel Motor Company of America in 1898; secretary of the Washington Company, contractors, New York; and consulting engineer, New York. Latterly he was associate professor of mechanical engineering at Armour Institute of Technology, specializing in thermodynamics, gas and oil engines, compressed air and refrigeration. He was the author of a number of technical papers, the owner of various patents, and had done special research work on boiler efficiency, gas engine efficiency, regenerator efficiency, etc.

He was an engineer in the best sense, a lover of truth, very careful in considering the different aspects of a problem, giving each due weight, and deciding each point after intensive study. He truly sank himself in his work, and presented his final conclusions with such modesty and amiability that a casual observer might easily have failed to realize the conscientious work and clear insight which prompted the conclusion. In the early work of the introduction of the Diesel motor into this country, these qualities of mind and the thoroughness of his equipment in mathematics and thermodynamics were of inestimable value; and were gratefully acknowledged by the late Dr. Diesel. His domestic life was for years a very happy one. His wife was a lady of many pleasant accomplishments, and a cheerful disposition, which enabled them to accept the ups and downs of an engineer's life with a noble equanimity. They had one child, a boy of rare promise, who combined the fine qualities of both parents. In his eleventh year he was run over by an automobile and instantly killed in the presence of his father, whose side he had left but an instant before. This was no doubt the cause of our friend's early death, to which the physician gives a technical name.

E. D. M.

### JOHN FRITZ

John Fritz died at his home, Bethlehem, Pa., on February 13, 1913, in the 91st year of his age. For over sixty years he was actively engaged in the upbuilding of the iron and steel industry, that period when practically every advance in the art took place which contributed to the great engineering achievements of the present day.

There is in the possession of The American Society of Mechanical Engineers a hand-turning tool used by John Fritz when learning

his trade, in place of the usual slide rest and fixed tool. In 1896, when President of the Society, Mr. Fritz in his presidential address reviewed the progress in the manufacture of iron and steel, and showed a full size drawing of the most modern type of heavy lathe of that day, weighing over 190,000 pounds. The drawing was much too long to be hung on one wall of what was then the auditorium of the Society, and in contrast with it the hand tool was shown, illustrating in a most striking manner the remarkable development in machine shop practice during the professional life of one man, and typical also of changes correspondingly great in every other department of the iron industry.

In his Autobiography, published last year, John Fritz begins the first chapter by saying, "I was born August 21, 1822, in Londonderry Township, Chester County, Pa., and was the oldest child of a family of seven children, three brothers and four sisters. I was born of parents of exemplary character, my father being a man of high moral standards; he fully impressed upon my mind the importance of absolute integrity, energy and economy. My mother was a true Christian woman . . . to my mind, the moral and religious training received from my parents was the most important training I could have possibly received; and I have ever thought the highest honor I could pay to their memory was to endeavor to follow their noble example."

His father was a millwright by trade and lived on a small farm, and John worked on the farm until sixteen years of age. This was before the days of public schools in that section and his education was obtained at district schools which kept three months in the winter and three months in the summer. The work on the farm, however, and the difficulties encountered in securing an education were but as endowments to his character which, combined with the sturdy stock of his ancestry, caused him resolutely to master the great problems of his later years. Along with these qualities which contributed to his success, were those other finer ones which made for him the many warm friendships both among his business associates and the workmen who were under him.

In 1838 he started to learn his trade as blacksmith and machinist. His most important early work was done at Norristown, Pa., where he went in 1844 to assist in building the Norristown Iron Works, and where he was soon placed in charge of all the machinery of the plant. This was a responsible position and an onerous one. There

were three sets of rolls in the mill, all driven by one engine, and so much trouble was experienced with the gearing that he became hostile to all geared mills and determined that if opportunity ever offered, he would build a mill direct-driven without the use of gears. In spite of the long hours and many difficulties encountered at Norristown, young Fritz determined to become proficient in all branches of the iron industry, and regularly spent his evenings until a late hour at work at the puddling furnace and in other departments of the mill.

In 1854 Mr. Fritz went to Johnstown, Pa., as general superintendent of the Cambria Iron Works. He was employed to remodel and rebuild the plant. The mill was of such a design and was so incomplete in many ways, that he found it impossible to make the rolling of rails in the ordinary way of commercial success. He therefore asked the proprietors to build a mill according to his long-cherished plan, one with three-high rolls, direct-driven, without gearing. This proposition met with the most strenuous opposition, resulting in Mr. Fritz giving an ultimatum to the effect that this step must be taken and that he would not consent to build a geared mill. Permission was at last granted and with the introduction of the three-high system of rolling and other improvements, the Cambria Iron Works became the greatest plant of the time.

The final period of Mr. Fritz's career began in 1860 with his removal to Bethlehem, Pa., to become the general superintendent and chief engineer of the Bethlehem Iron Company, and to design and erect the company's plant. Three years later the rolling of rails was commenced and the plant was universally recognized to be remarkable in its conception and design. Here there were successive developments, which Mr. Fritz directed until his retirement. Among them were the Bessemer plant, the open-hearth process, the 125-ton steam hammer, the largest in the world, intended to produce more perfect forgings than possible with smaller hammers, the Whitworth fluid compression process with its 14,000-ton hydraulic forging press, operated by 15,000-horsepower engine and pumps, and the massive tools required for machining the forgings, and the armor plate plant for making steel plates by the Creusot process. The importance of this and similar forging plants in this country in making possible the heavy types of machinery which were constructed in the latter part of the century, such as the power units at Niagara Falls, and

the large steam engines which immediately preceded the development of the steam turbine, cannot be overestimated.

Besides his active work with the various companies with which he was connected, Mr. Fritz designed for the Government a rail rolling mill erected at Chattanooga, Tenn., during the Civil War, and made plans and specifications for an armor plant for the Government, which, however, was not constructed.

Mr. Fritz received many high honors from various engineering societies and educational institutions. He was president of the American Institute of Mining Engineers in 1894, president of The American Society of Mechanical Engineers in 1896, and an Honorary Member of this Society and of the American Society of Civil Engineers. He received the Bessemer gold medal of the Iron and Steel Institute of Great Britain and was elected Honorary Member of that Society. Honorary degrees were conferred on him by Columbia University, the University of Pennsylvania, Stevens Institute of Technology and Temple University.

Perhaps no greater mark of esteem has ever been bestowed on an engineer than that tendered Mr. Fritz in commemoration of his seventieth birthday. A banquet was proposed by members of the Engineers Club of New York, at first with the idea of a small affair. As the plans developed, however, 300 friends and associates sought the privilege of attendance, among whom were distinguished representatives of the Government and of many institutions and firms, necessitating the use of the Grand Opera House at Bethlehem for the dinner. It was one of the most notable meetings in the annals of American engineering.

Ten years later, at the time of his eightieth birthday, a banquet was held in Mr. Fritz's honor at the Waldorf-Astoria in New York which also signalized the founding of the John Fritz medal by members of the four national engineering societies, a permanent fund being contributed for the annual purchase of the medal. In this way his friends have assured the perpetuation of the memory of his achievements in industrial progress.

When Lehigh University was founded in 1866, Mr. Fritz was selected as one of the original trustees and he was deeply interested in the success of the institution, as evidenced by the establishment through his generosity of the John Fritz engineering laboratory; and in his will there is provision for the further endowment of this institution.

He further made a fitting memorial to his parents in the erection at South Bethlehem of the Fritz Memorial Methodist Episcopal church and parsonage.

Of Mr. Fritz's family there survive only two sisters, a niece and three nephews. His wife died in 1908, and their only child in 1860.

The life of Mr. Fritz constitutes one of the most notable examples of an American born of poor parents and with limited education who achieved by his own personal qualities the highest success, not only in material things but in reputation and a wide circle of friends.

### LUDWIG HERMAN

Ludwig Herman was born in Prague, December 2, 1842, and died in Cleveland, Ohio, on October 21, 1913. He received his education at the University of Prague and in the machine shops of Vienna. In 1865 he came to America, residing first in New York, where he was employed by the Brush Electric Light Company and the Yale & Towne Manufacturing Company, and later in Chicago, where he became an engineer for the Rust Bridge Company. He was for several years chief engineer of the Detroit Bridge & Iron Works Company, in Detroit, Mich., and later became a member of the firm of the Fox & Hower Bridge Company of Chicago, designing and building numerous bridges all over the country, among them that over the Mississippi River at Hannibal, Mo., one of the largest bridges of its time. In 1885 Mr. Herman became general manager and engineer of the Buckeye Machine Company of Cleveland, Ohio.

Mr. Herman designed the first 20,000 candle power arc lamp for the Paris Exposition, and lowered the first electric light mast in 1885 at Akron, Ohio. He planned and executed the structural iron work of the tower of the Fairmount Street reservoir and executed the structural iron work for the additional stories and roof of the old court house of Cleveland.

He was a member of the Cleveland Engineering Society.

### VICTOR HUGO

Victor Hugo was born at Kingston, Ontario, November 20, 1873, and received his engineering education at the University of Minnesota, from which he was graduated in 1896 with the degree of M. E. He began his career as assistant to his father, T. W. Hugo, consulting mechanical engineer of Duluth, in July 1896, and two years

later entered the Chicago inspection force of the Hartford Company. Late in the following year he was transferred to St. Louis, and shortly afterwards was appointed chief inspector of that department. On January 1, 1905, he was promoted to the position of manager, a position he held till his death on January 31, 1913.

### GEORGE T. INGERSOLL

George T. Ingersoll was born at Schenectady, N. Y., November 6, 1847. After receiving a public school education, he entered the Schenectady Locomotive Works as an apprentice machinist, where he remained four years. He then secured a position in the Illinois Central Railroad shops at Chicago, and after three years returned East to assist in the reconstruction of the shops of the New York Central Railroad at Syracuse. When this work was completed his company transferred him to West Albany. In 1876 Mr. Ingersoll was appointed under-sheriff of Schenectady County, a position which he filled for three years. Until 1883 he was engaged in delivering engines at the Schenectady Locomotive Works when he was made foreman of the company. In 1899 he was appointed superintendent of the Schenectady Water Works, a position he held for ten years. Mr. Ingersoll then retired from active life, and died at Worcester, N. Y., November 10, 1913.

### PETER KIRKEVAAG

Peter Kirkevaag was born in Christianssund, Norway, April 1849, and in 1871, after finishing an apprenticeship, went to Germany with a stipend from the Norwegian government. He was graduated from the polytechnic school in Langensalza, Turingen, in 1874 and was afterwards employed as draftsman and engineer in Westphalia until 1877, following which he was inspector for three years for the Nordenfelth gun factory in Stockholm, Sweden. In 1881 he came to the United States, where he secured employment as machinist and draftsman with Oliver Brothers & Phillips and A. Garvison & Co., Pittsburgh, Pa. Two years later he became draftsman and superintendent of buildings, foundations, etc., for the Hartman Steel Company, Beaver Falls, Pa., and this same year saw the beginning of his connection with William Tod & Company, Youngstown, Ohio, which extended over many years. At the time of his death on May 6, 1913, he was in the employ of the Brier Hill Steel Company of Youngstown.

### WILLIAM H. LEACH, JR.

William H. Leach, Jr., was born in Brooklyn, N. Y., September 4, 1873. He received his technical education as a mechanical engineer at the Massachusetts Institute of Technology, being graduated in 1897. Immediately thereafter he entered the employ of the Union Metallic Cartridge Company of Bridgeport, Conn. During his fourteen years of service there he acquired an intimate knowledge of many of the manufacturing processes used in the fabrication of the companies' products and designed machines for manufacturing primers and for packing them, for loading and packing cartridges, for weighing powder charges and for delivering them into their cartridge shells, loading apparatus for rapid-fire ammunition, etc. His work consisted also of laying out and setting up machines, and systematizing the work in the engineering department. During the last two years of his business life he was research engineer for the company and was considered one of the foremost experts on ammunition in the country. He died in Newton, Mass., September 6, 1913, after an illness of two years.

### FRANCIS VALENTINE TOLDERVY LEE

Francis V. T. Lee was born at Winchester, Hampshire, England, August 28, 1870, and attended the grammar school at Manchester, England, and the College Communal at Boulogne, France. He came to Sherbrooke, Canada, in 1887, and was for several years in the service of the Canadian Pacific Railway as secretary to the chief of construction, in charge of the forwarding of material. After a year in England he came to New York and entered the employ of the Manhattan Electric Light Company, as assistant under E. E. Stark. Here he came into contact with the late F. A. C. Perrine and formed the acquaintance that so strongly influenced his career.

This acquaintanceship led him to take up the study of electrical engineering at Leland Stanford Junior University, under Dr. Perrine, then professor of electrical engineering there, to whom he acted as secretary and general laboratory assistant. Shortly after graduation from the university in 1897, he was appointed assistant engineer to John Martin, agent for the Pacific Coast department of the Stanley Electric Manufacturing Company. He rose rapidly in this service, and in 1900 was made vice-president and general manager of John Martin & Company, electrical engineers and contractors, and also district Pacific Coast manager for the Stanley Electric Manufactur-

ing Company, and many other Eastern manufacturers. During this period there came under his direct supervision the erection of many of the earlier lighting and power plants that later were absorbed by the Bay Counties Power Company and the Pacific Electric Railway Company.

Early in 1906 he severed his connection with John Martin & Company, but followed Mr. Martin's interest into the Pacific Gas & Electric Company, where he was made assistant to the president; as such he was generally responsible for the construction and operation of the hydroelectric developments of that company. In 1910 he resigned from the service of the Pacific Gas & Electric Company, returning to his old home in England for a visit and spending some time in travel on the Continent. He died in Victoria, B. C., on August 17, 1913, shortly after his return to America. He was a member of the American Institute of Electrical Engineers, of the American Society of Civil Engineers, of the American Gas Institute, of the American Electrochemical Society, and of the Institution of Electrical Engineers in England.

### WILLIAM MASON

William Mason who until his retirement from business a few years ago was master mechanic of the Winchester Repeating Arms Company, of New Haven, died in Worcester, Mass., on July 17, 1913. He was born in Oswego, N. Y., on January 30, 1837, and early showed his natural taste for mechanics. His training for his profession was obtained through apprenticeship as a patternmaker and machinist, his first connection being with the Remington Arms Company at Ilion, N. Y. After a long association with this company he resigned to enter the Colts patent firearms works at Hartford, and later the Winchester Arms Company.

Mr. Mason was the inventor of many appliances for looms and weaving, steam pumps, bridge work, and for arms and ammunition and the machinery connected with their manufacture, and also assisted in the design and construction of the Knowles steam pump and Knowles fancy looms. He was a member of the Union League Club, New Haven, and also of a number of scientific societies.

### HERMAN CHARLES MEINHOLTZ

Herman Charles Meinholtz was born February 7, 1868, in St. Louis, Mo., and died in the same city on December 24, 1913. Mr.

Meinholtz was educated in the public schools of his native city and in 1883 entered the Manual Training School of Washington University, graduating in 1886. After a short service in the university's testing laboratory and later as timekeeper in the Shickle, Harrison & Howard Iron Works, he became an assistant draftsman for the Heine Safety Boiler Company of St. Louis. The company was then having its boilers built by contract and young Meinholtz was sent to inspect and accept the work. His keen observation, straightforwardness and courage were manifest in this service and brought him rapid advancement.

In 1892 he was made chief draftsman and in 1895 superintendent; in 1899 he was placed in charge of the company's first shop in St. Louis which he gradually improved and enlarged. In the meantime the Phoenixville shop was being developed and he utilized this opportunity for comparison when made vice-president in 1907, and worked out a design for a special Heine boiler shop which was built in 1909 at St. Louis and stands as a monument to his ability in practical design.

Mr. Meinholtz was a member of the Engineers Club of St. Louis. Since 1911 he had been a member of the Society's committee to Formulate Standard Specifications for the Construction of Steam Boilers and other Pressure Vessels, where his practical good sense and exact knowledge found full scope. His twenty-six years as designer and constructor securely fixes his work in the profession.

E. D. M.

## SAMUEL EDWARD MITCHELL

Samuel Edward Mitchell was born at Halifax, in June 1875, and received his education at the Halifax Higher Grade School and at the Halifax Technical School. In 1892 he was apprenticed with Fred Hanson & Company, machine tool builders of Halifax, in the works on machines, and two years later went to the Campbell Gas Engine Company, also of Halifax, gas and oil engine builders. He remained two years in the works here and then was promoted to the drafting room, where he remained until 1902, having risen to assistant chief draftsman when he left. In August 1902, he became chief gas engine designer for Tangyes, Ltd., in Birmingham, England, and remained there until April 1905, when he became head of the gas engine department of Ruston, Proctor & Company of Lincoln, England. After two years spent with this company, he came to America and became identified with the Jacobson Engine Company of Chester, Pa., where

he remained two years as superintendent, building engines up to 400 h.p. In 1909 he spent nearly a year as gas engine designer with the Minneapolis Steel & Machinery Company, of Minneapolis, Minn., on their large double-acting gas engines. From July 1910, until his death he was general superintendent of the Geo. D. Pohl Manufacturing Company, of Vernon, N. Y., builders of gas and gasolene engines, and he had just completed designs of a new line of gas engines for this company and was engaged in some important developments in oil engines at the time of his death, August 19, 1913.

### EVERETT FLEET MORSE

Everett Fleet Morse, inventor of the Morse chain for bicycles and transmission of power, was born June 28, 1857, at Ithaca, N. Y., and died November 11, 1913. He received his early education in the public schools of Ithaca, and was graduated in the class of 1884 from Sibley College, Cornell University. From 1884 to 1890, he worked on the design of an automatic horse-drawn hay rake, perfected inventions in carriage springs, and also worked as solicitor of patents. He was the inventor of the Morse thermo gage, an instrument for measuring the temper of steel, which had been adopted by the United States and German Bureaus of Standardization. Mr. Morse took a profound interest in public affairs and welfare, and was recently elected president of the Ithaca Hospital Association. He was also a member of the building committee of the new Ithaca City Hospital.

### SAMUEL LYON MOYER

Samuel Lyon Moyer, first vice-president of The Lunkenheimer Company of Cincinnati, Ohio, died at his home in that city on May 3, 1913. Mr. Moyer was born in Cincinnati on August 17, 1874, and was educated in the public schools. He had been connected with the company since 1890, almost from the time of the completion of his school work, working his way up to the management from a small beginning, entirely by his industry and rare ability.

He was a deep student of men and affairs, possessing clear vision and remarkable foresight, and contributed at every opportunity of his time, money and intellect for the growth and advancement of his native city. While not an active engineer he was well informed and keenly interested in engineering and scientific tests of the day, particularly in the line of progress and development.

Mr. Moyer was a member of the National Metal Trades Association, and a number of social and business clubs.

### EDWARD J. MURPHY

Edward J. Murphy was born in the province of Ulster, Ireland, February 5, 1829, and was educated in the private schools of Dublin, his training being that of a civil engineer. In 1849 he came to America, and the following year made surveys in Ohio for a Philadelphia map publishing company, and later did the same kind of work in the central part of New York. In 1853 he became first assistant to the city surveyor of New York, helping to lay out many of the street car lines of the city. Two years later he went to Hartford as chief draftsman for the Woodruff & Beach Iron Works, and in this capacity was connected with some of the most important work for the government during the Civil War, being identified in the designing and constructing of the machinery for the United States sloops of war Mohican and Kearsarge and also with the gunboats Pequot and Nipsic, and later with the three large sloops of war Piscataqua, Minnetonka and Manitou.

After the war the Nelson Mining Company was organized in Hartford and Mr. Murphy went to Montana to look after the interests of Woodruff & Beach, in the hope that the change would improve his health. Soon after his return the Hartford Foundry & Machinery Company was formed to succeed the old firm of Woodruff & Beach, and Mr. Murphy was made secretary and treasurer of the company at its organization in 1872. In 1878 he was elected president of the Board of Water Commissioners, from which he resigned two years later to accept an appointment of superintending engineer at the factory of the Colt Patent Fire Arms Manufacturing Company. He remained with this firm until 1889, when he resigned to become the consulting engineer of the Hartford Steam Boiler Inspection & Insurance Company. He died September 2, 1913.

Mr. Murphy was honored with many places of public and private trust during his long residence in Hartford. He was one of the charter members of the Society and an associate member of the American Society of Naval Engineers.

### ALEXANDER ZECK NEWLIN

Alexander Zeck Newlin, mechanical engineer of the National Tube Company, Kewanee, Ill., was born at McKeesport, Pa., on October

17, 1876, and received his education in the public schools of that city. From 1896 to 1898 he taught school, taking meanwhile a special correspondence course in mathematics at the University of West Virginia, Morgantown, W. Va. In 1898 he entered the drafting room of the National Tube Company at McKeesport, and worked his way up through apprenticeship in the galvanizing department to the engineering department. At the time of his death on October 7, 1913, he was in charge of the design of valves and fittings and of the design and installation of new equipment throughout the Kewanee plant.

### EDGAR PENNY

Edgar Penny, for many years connected with the ice machine and steam engine industry in Newburg, N. Y., as well as the work of the Alberger Condenser Company, died in Auburn, N. Y., November 14, 1913. He was born in Port Jervis, N. Y., February 13, 1845, and came to Newburg when a boy to learn the trade of machinist in the Highland Iron Works. At the beginning of the Civil War he enlisted in the United States Navy and saw service in the South, continuing at the close of the war as assistant engineer and visiting Japan, China and other foreign ports. After three years he returned to Newburg to take up his work with the Whithill Engine and Pictet Ice Machine Company, resigning a few years later to become managing director and mechanical engineer of the Frick Company, Waynesboro, Pa. An ice machine invented by Mr. Penny, which was known as the "Frick" or the "Eclipse," was built by this company.

In 1895 he returned to Newburg and formed the Newburg Ice Machine Company, of which he became the first president. A few years ago when the late Louis R. Alberger bought out the Newburg Ice Machine and Engine Company, known later as the Alberger Pump and Condenser Company, Mr. Penny became its vice-president, retaining at the same time his position as general manager of the old company. Recently his duties had been lightened when he was made consulting engineer of his company.

### JOHN BRADFORD PERKINS

John Bradford Perkins was born in Boston, August 2, 1869. He was educated in the public schools of Lowell, Mass., and served an apprenticeship with the General Electric Company of Lynn, Mass., and the Lowell Machine Company of Lowell. Upon its completion

he entered the employ of Hollingsworth & Vose, at Walpole, Mass., resigning to enter the Crosby Steam Gage & Valve Company. In 1905 he became the New England manager for the Hewes & Phillips Iron Works, Newark, N. J., manufacturers of Corliss engines. The John B. Perkins Company, a firm which engaged in the installation of complete power plants, was organized by him the following year, Mr. Perkins serving as its president up to the time of his death on January 19, 1913. In this capacity he made many important installations, notably the Fitchburg Yarn Company, with its record for economical operations, the Windham Manufacturing Company, Willimantic, Conn., the Burgess Mills, Pawtucket, R. I., and the Potomsko Mills, Fall River, Mass. He also designed and put on the market the "Bradford" valves.

Mr. Perkins was a member of the Engineers' Club of Boston, the New England Street Railway Club, the National Association of Stationary Engineers, the Nayassett Club of Springfield, the Quequeehan Club of Fall River, and the Deerfield Club of Manchester, Vt.

### THURSTON MASON PHETTEPLACE

Thurston Mason Phetteplace was born at Providence, R. I., on May 3, 1877, and received his education there, in the English and Classical High Schools, and in Brown University, where he was graduated in 1901 in mechanical engineering. After leaving school he was employed at the Builders Iron Foundry and as instructor in drawing at Brown University, later becoming instructor in mechanical engineering there. In 1909, he received a degree of Master of Arts from Columbia University, and was appointed associate professor of mechanical engineering at Brown University. At the time of his death on September 7, 1913, he was engaged, in addition to his professorship, in consulting work in the firm of Kenerson, Brooks & Phetteplace, his specialty being automobile construction and gas engine design. He was president of the Providence Association of Mechanical Engineers, and held membership in the Society for Promotion of Engineering Education.

### FRANCIS M. RITES

Francis M. Rites was born at Petersburg, Ill., on July 20, 1858, and received his preparatory education at Chester, N. Y. He entered Sibley College, Cornell, in 1877, graduating with the degree of B.M.E. in 1881.

Directly after leaving college, he entered the employ of the Lehigh & Hudson River Railroad for a short time, and in 1883 became identified with the Westinghouse Machine Company. During his connection with them he made many notable developments and inventions, among them a system of high-speed compressors, a new system of explosive engine control and distribution, and the inertia governor, for which he is best known. He died at his home in Slaterville, N. Y., on May 2, 1913.

### ELMER A. SAMMONS

Elmer A. Sammons was born in Cheboygan, Mich., March 31, 1860, and at the early age of fifteen was made captain of the steamer Minnie Sutton, on Lake Superior. In 1880 he moved to Cincinnati, where he carried on a general consulting business for the following firms: George Enger Carriage Company, Miller Plating Works, Union Laundry Company, T. A. Snyder Preserve Works, Henry Geiershoffer Clothing Company, and W. H. Meredith and Company, general machinists. He conducted a night school from 1885 to 1888, in connection with the Marine Engineers Beneficial Association, of which organization he was president and vice-president. In 1888 Mr. Sammons went to Louisiana where he took charge of the sugar refinery of James H. Laws & Company at Cinclare, and three years later he remodeled the entire plant, installing the revolving grate bar, one of his own patents, which proved to be so successful that it was put into practically every large factory throughout Louisiana. During this time he also superintended the erection of the A. Wilbert Refinery at Plaquemine, La. Mr. Sammons moved to New Orleans in 1903 and engaged in the general machinery business, one of his greatest undertakings being in connection with the sugar refinery at Thibodaux. He died at New Orleans, July 4, 1913.

### ANDREW MARTIN SCHREUDER

Andrew Martin Schreuder was born in Syracuse, N. Y., October 17, 1874. He obtained his early education in the public schools of that city and his technical training at Cornell University, where he was graduated in 1897. He was subsequently connected in various capacities with the Geo. M. Newhall Engineering Company, the Tabor Manufacturing Company and the Woodbridge Manufacturing Company. In 1908 he became associated with the Philadelphia Textile Machinery Company, at the time of his death on January 1, 1913,

being its superintendent and chief engineer. He had developed designs for and successfully built and installed a number of the largest dryers in operation and was engaged in the development of dryers for explosives and inflammable materials.

He was a member of the Cornell Club and of the Franklin Institute of Philadelphia.

### OLIN SCOTT

Olin Scott was born February 27, 1832, at Bennington, Vt., where he learned the trade of a millwright. In 1858 he formed a partnership with H. S. Brown and established the Bennington Machine Works; in 1863 he purchased the interests of his partner and in 1864 purchased and combined the business of the Eagle Foundry and Machine Works. Later this foundry developed into a factory for the manufacture of powder mill and pulp mill machinery. Before the use of nitro powders became general, the Bennington Machine Works had acquired a national importance and worldwide reputation, machines from its shops having been shipped to every continent on the globe. After the close of the Civil War Colonel Scott ceased to confine his energies exclusively to manufacturing and became an organizer of powder manufacturing companies. In 1869 he built the Lake Superior Powder Mills at Marquette, Wis., and four years later he became superintendent of the Laflin and Rand Powder Company of New York. In 1882 he organized the Ohio Powder Company of Youngstown, Ohio, and for several years was vice-president of the corporation. He also organized the Pennsylvania Powder Company at Scranton, Pa., in 1884, becoming its president. Three years later he disposed of his interests in Ohio and Pennsylvania and became a consulting engineer for the Laflin and Rand Company and the Dupont Powder Company of Wilmington, Del., a position he retained until 1894. In 1892 he became president of the Lasher Stocking Company and operated the property until its comparatively recent disposition to the Vermont Hosiery and Machinery Company.

Colonel Scott died in Bennington, April 28, 1913.

### HAROLD SERRELL

Harold Serrell was born in Brooklyn, N. Y., August 26, 1852. Having completed his education at Adelphi Academy in 1869, he entered the office of his father, Lemuel Wright Serrell, patent attorney and expert in patent causes, with whom he was associated under the

firm name of L. W. Serrell & Son. After his father's death in 1899, he carried on the business alone, studying and familiarizing himself with machinery, mechanical devices and the arts and sciences for the professional career of solicitor of patents and mechanical expert. He served in contested cases as mechanical expert, giving testimony both in court and before a master for use in court. He died February 26, 1913.

### DANIEL SIMONDS

Daniel Simonds, president of the Simonds Manufacturing Company of Fitchburg, Mass., the largest saw manufacturers in the world, and a man widely known in industry, died at his home in Larchmont, N. Y., on May 5, 1913.

Mr. Simonds was born in Fitchburg on September 18, 1847, and received his education in the common schools there and at Comer's Commercial College in Boston. He started work with his father, Abel Simonds, in West Fitchburg, making scythes and edge-tools.

From a modest beginning the business of the Simonds Manufacturing Company has grown to become one of the big industries of the country. In 1868 the Simonds Manufacturing Company was incorporated and the plant moved to Fitchburg. The new company took over the business organized by Abel Simonds, and Daniel Simonds held various offices in the company from time to time as the company grew.

In the early eighties Mr. Simonds was made superintendent and vice-president of the company and in 1888 he succeeded George F. Simonds as president. Under his leadership the business branched out, acquiring plants at Chicago, Montreal and Lockport, N. Y., and in 1905, a new and enlarged plant was constructed on the site of the old one at Fitchburg at a cost of a quarter of a million dollars. There are many branch offices in the United States, as well as London, England; St. John, N. B.; and Toronto, Ont. The Simonds Manufacturing Company has always been most liberal towards its employees and has several organizations which have been of direct benefit to them. Mr. Simonds was the father of the coöperative industrial system of education at the high school in Fitchburg, which through his personal interest has attained great success.

### COLIN C. SIMPSON

Colin C. Simpson, assistant secretary and general superintendent of mains of the Consolidated Gas Company, died in New York, April

8, 1913. Mr. Simpson was born at Maidstone, England, on December 16, 1856, and received his education in private schools, the Technical High School of Gratz, Austria, and the Naval Academy at Trieste, Austria. He began his career with an English contracting firm laying water mains in Vienna, and during his connection with them invented a machine for tapping mains while under pressure without allowing water to escape.

In 1880 he came to the United States and entered the engineering department of the Knickerbocker Gas Company. Two years later he was placed in charge of mains of the Municipal and Knickerbocker Gas Companies, and at the time of their consolidation in 1884 into the Consolidated Gas Company, was made district superintendent of mains. From this he rose to the general superintendency and also to the position of assistant secretary, in both of which he continued up to the time of his death.

Mr. Simpson had an unusual acquaintance with sub-surface conditions in New York City, having had charge of the design and construction of all gas mains laid in Manhattan since 1884. He designed and successfully laid during 1910 and 1911 two 36-inch and one 48-inch main across the Harlem River, one of the most difficult problems in connection with main work. He also laid the East River Gas Company's 60-inch steel main from the Astoria Works to Ravenswood, the largest gas main in the world. He was the first man to use lead wool for pipe jointing in this country, having imported this material from Germany before it was manufactured here. He was a leading exponent of absolute safety with respect to gas mains, and designed and secured the adoption of the system of bypassing of all gas mains, now in general use during the construction of the subways. Mr. Simpson had a very wide acquaintance among his profession and was frequently called upon as an expert in damage suits against gas companies. He appeared before the United States Supreme Court in the New York Eighty-Cent Gas Case and established the value claimed for the mains under his charge. He was a member of the American Gas Institute, the Engineers Club, the Society of Gas Engineering, the Society of Illuminating Engineers, and the National Democratic Club.

### GEORGE HAMPTON SMITH

George Hampton Smith was born in Pittsburgh, Pa., September 29, 1878. He attended the public schools of Pittsburgh and imme-

diately after graduation from high school entered the Carnegie Steel Company where he was under the direct supervision of James Scott, superintendent of the Isabella and Lucy furnaces of the Carnegie Steel Company, Pittsburgh. His experience covered all phases of blast-furnace work, including the drafting room, and in this manner he gained a wide and thorough knowledge of blast-furnace construction. He was graduated in 1910 from the Carnegie Technical Schools in mechanical engineering and in 1911 was made assistant to the chief engineer under Mr. Scott at the steel works. In this capacity he had charge of practically all the construction and drawing room work in connection with the building of modern thin lined water-cooled furnaces with which he had great success. At the time of his death, May 20, 1913, he was engaged in remodeling the stacks of the Carnegie company's Edgar Thompson works at Braddock, Pa.

Mr. Smith was a member of the Railway Club of Pittsburgh, and president of the Council of Aspinwall, where he had made his home for the past ten years.

### FREDERICK P. THORP

Frederick P. Thorp was born at Wyandotte, Mich., September 2, 1868, and after an education in the public schools and at the Chicago Manual Training School, entered the North Chicago Rolling Mills in South Chicago as a machinist. In 1889 he was employed in the shop of the Brush Electric Company, Cleveland, Ohio, and from 1890 to 1898, served in the drafting room of the Detroit Electric Works and the Siemens & Halske Electric Company.

In 1902 he entered the employ of the Westinghouse, Church, Kerr Company, New York, as a salesman, leaving there to become associated with the Power & Mining Machinery Company, New York, where he remained for ten years. On August 1, 1913, he resigned to engage in engineering work for the State of New York, with which he was occupied up to the time of his death on December 21, 1913.

### CHARLES E. TOMLINSON

Charles E. Tomlinson was born February 14, 1868, in Auburn, N. Y., and was educated in the public schools of that city and of Syracuse. He entered the field of mechanical engineering by serving as an apprentice with a number of firms, including the L. C. Smith Gun Company, LaFevere Arms Company, Weston & Smith and I. Weston & Company, drafting machinists, designers and builders,

Duell, Laass & Duell, Emil Laass & Company, Hey, Wilkinson & Parsons and Hey & Parsons, patent solicitors and attorneys, all of Syracuse, N. Y. In 1900 he entered the employ of the Remington Typewriter Company as designer and draftsman, and in 1903 transferred to their Smith Premier plant, with which he was connected at the time of his death, on March 10, 1913. Mr. Tomlinson designed and constructed the various parts of the typewriter for the Remington Company, and during the latter years of his life was occupied mainly in expert work on patents and related questions.

### ARTHUR PIERCE TRUETTE

Arthur Pierce Truette was born in Boston, Mass., December 28, 1888, and died in Brookline, Mass., on December 17, 1913. He was educated in the Allston Grammar School of Boston and at the Brookline High School, and was graduated from the Massachusetts Institute of Technology in 1910. He acted as assistant at the Institute for one year and in August 1911, entered the service of the Goodyear Tire and Rubber Company, of Akron, Ohio, as power plant engineer, remaining with them until shortly before the time of his death. During his brief connection he reorganized and doubled the capacity of the company's power plant.

Mr. Truette was a member of the University Club, the Goodyear Technology Club and the Chamber of Commerce of Akron, the Northern Ohio Technology Club, the New York Technology Club, and the Appalachian Mountain Club of Boston.

### AARON VANDERBILT

Aaron Vanderbilt of Remsenburg, New York, died on March 25, 1913. He was born in Brooklyn, N. Y., January 29, 1844, and was educated in the public schools of Staten Island and Brooklyn. Throughout his professional career he was connected with marine work and had no technical training other than that gained by experience.

He was at one time manager of the Ward Steamship Company, and at the time of the formation of the Society was superintendent of the New York and Cuba Mail Steamship Company. From 1900 to 1908 he was vice-president of the Wheeler Condenser & Engineering Company, ill health forcing him to resign at the end of this period. He was especially interested in marine engines and gave much time and study to their development.

During the Civil War while on Admiral Porter's staff he drew plans of the enemy's fortifications at Fort Fisher and elsewhere.

Mr. Vanderbilt was elected to life membership in the Society in 1908. He was a member of the United States Naval Institute, the Naval Order of the United States, the Society of Marine Architects and Naval Engineers, the Navy League, the Grand Army of the Republic, and the military order of the Loyal Legion. He was largely instrumental in the formation of the Naval Reserve and the Navy League, and was at one time chairman of the committee on ocean transportation of the New York Board of Trade.

### WILLIAM OLIVER WEBBER

William Oliver Webber was born at Springfield, Mass., January 4, 1856, and was educated in the public schools of Manchester and Portsmouth, N. H., and of New York. He served his apprenticeship at the Manchester Locomotive Works from 1871 to 1874, his subsequent experience covering engineering work in the locomotive department of the Boston & Maine Railroad Company; with the Toledo, Peoria & Warsaw Railroad Company, Peoria, Ill.; with the United States Centennial Exposition, Philadelphia, in 1876, and in the test department and laboratory of the Chicago, Burlington & Quincy Railroad Company, Aurora, Ill. In 1881 Mr. Webber was made superintendent of the Allen Paper Car Wheel Company, Pullman, Ill., and six years later, commissioner of Abolition of Grade Crossings in Massachusetts. He relinquished this position one year later to accept the superintendency of the Erie City Iron Works. Since 1894 he has been a consulting engineer in the firm of Webber & Smith, Mechanical, Hydraulic and Dynamic engineers, of Boston. He acted as expert in the United States and Circuit Courts.

Mr. Webber had been a member of the Ancient and Honorable Artillery Company, Massachusetts, since 1884, and engineer officer of the First Brigade Staff, Massachusetts Volunteer Militia, from 1901 to 1904. He was a member of the Boston Society of Civil Engineers, the Society of Colonial Wars, United States Military Institute and the United States Naval Institute. He died January 9, 1913.

### SIR WILLIAM HENRY WHITE

The distinguished Naval Architect and Honorary Member of The American Society of Mechanical Engineers, Sir William Henry White,

passed away on the 27th of February, 1913, bearing the profound respect of the engineering profession throughout the world.

He entered the service of his country as an apprentice, and by assiduous application and inborn ability rose to the highest rank in naval construction in the British Admiralty.

Born in 1845, he entered the British Dockyard at Devenport as an apprentice at the age of fourteen. While working in the shops he attended the dockyard school and won the Admiralty Scholarship in 1863. During the year 1864 the Royal School of Naval Architects was established at South Kensington. Young White took first place in the first entrance examinations of that school, maintained first place, and was graduated first with the honorary degree of Fellow in 1867. He immediately entered the British Admiralty and remained there until 1902, when he was obliged to resign by reason of failing health.

In 1873 he became Secretary of the Council of Construction of the Navy under the Presidency of Sir Nathaniel Barnaby. He rapidly rose to the rank of Chief Constructor when in 1881 he resigned to take charge of the shipyard newly organized by Sir William Armstrong at Elswick, England. Here he designed many warships for foreign governments, including two cruisers for the United States. Upon the resignation of Sir Nathaniel Barnaby in 1885, and upon the recommendation of his former chief, he was recalled to the Admiralty as Director of Naval Construction.

At this time the question of largely expanding and rebuilding the British Navy was under consideration, and he found himself in charge of the engineering of this great work. Continuously for seventeen years, and until his health was broken by overwork in 1902, he labored on and revolutionized the navy. During that period he designed 245 warships, including 43 battleships, 202 cruisers of different classes, and many torpedo boats and destroyers, which were built at a cost exceeding $500,000,000. The largest warship previous to this time was 340 feet long and 10,600 tons displacement. His ships of the King Edward VII class were 425 feet long and 16,350 tons displacement. The speed of armored cruisers during his régime increased from 17 to 24.5 knots, the length from 315 to 500 feet, and the displacement from 8400 to 14,000 tons. His designs were so scientifically and accurately worked out that in no case did the actual ships exceed in draft or displacement the estimates of the design.

High propulsive efficiency was always realized, and in no instance did a ship fail to attain the required speed.

Mr. White was rewarded for his notable achievements by a C.B. in 1891, by a K.C.B. in 1895, and by a special grant by Parliament in recognition of "exceptional services to the Navy."

After his retirement from the Admiralty in 1902, he regained his health and was enabled to take up other important engineering work. He was one of the Cunard Commission that settled the question of propelling the Lusitania and Mauretania by steam turbines, and was a director of the firm of builders of the latter. He also designed steamers with geared turbines for service in India.

Sir William was distinguished as an author. His Manual of Naval Architecture is a classic, and of no less value is his Treatise on Shipbuilding. His many papers on many different engineering subjects presented before many engineering and scientific societies all contributed to his renown.

He was greatly interested in the education of the engineer and did much to elevate the standards of technical schools. His lectures in the Royal School of Naval Architecture from 1870 to 1881 resulted in the accession to the navy and private shipbuilding works of a new and much needed class of designers.

He was honored by many societies by election to offices of distinction. He was Honorary Vice-President of the Institution of Naval Architects; Fellow of the Royal Society; Honorary Member of The American Society of Mechanical Engineers, of the American Society of Civil Engineers and of the American Society of Marine Engineers and Naval Architects. He has been President of the Institution of Civil Engineers, the Institution of Mechanical Engineers, the Institution of Naval Architects, the Institution of Marine Engineers, the Institute of Metals, and was President-Designate of the British Association.

He received the honorary degrees of D.Sc. from Cambridge University and Durham, England, and from Columbia University, New York City, also of D.Eng. from Sheffield and LL.D. from Glasgow. In 1911 he was awarded the John Fritz Medal for "notable achievements in naval architecture."

Sir William White's personal character was known of all engineers. He was above all, straight and manly, a gentleman ready to see the good in others, yet vigorous and steadfast in his own convictions. He was intolerant of shams and of opinions based on self-interest, but

always ready to encourage the young engineer who was true to his profession.

Services were held simultaneously at Holy Trinity Church, Roehampton, and at St. Margaret's, Westminster, London.

The notable engineering societies and many other scientific and other organizations were represented at the services, including this Society. The interment is in the cemetery at Putney, London.

His wife, one daughter, and three sons, officers in the British Navy, remain to mourn his loss.

J. M. S.

### HENRY SHOTWELL WOOD

Henry Shotwell Wood, one of the foremost consulting engineers on hydraulic dredging machinery in the country, was born in Philadelphia, Pa., December 18, 1860. He was a graduate of Swarthmore College in civil engineering in the class of 1883, and until 1885 acted as inspector of drawbridge machinery for the Pennsylvania Railroad. From 1887 to 1906 he was designer and inspector of a special contractors' plant for the San Francisco Bridge Company, at the same time designing and building hydraulic dredges for the New York Dredging Company and the Atlantic, Gulf and Pacific Company. Among the largest dredges that he designed and built were the 23-inch suction dredges, Manila and Geo. W. Catt, the 22-inch dredges, Washington and Julia, and the 20-inch dredges, Florida, Port Royal, Texas City, Champlain and Fort Edward. He was the engineer and eastern agent of the North American Dredging Company of California.

Mr. Wood was a member of the American Society of Civil Engineers. He died December 5, 1913.

Index photographed at the beginning for the convenience of the microfilm user.